

NASA/MSFC Contract NAS 8-11303

NASA/MSFC Control No. DCN-1-4-50-01126, S1(1F)

CPB 02-1177-64

Solar Reference: SO 6-1612-7

## 1st QUARTERLY REPORT

# LIQUID HYDROGEN FLEXIBLE DUCTING TECHNOLOGY

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# 1st QUARTERLY REPORT

PERIOD COVERED 15 July through 31 October 1964

## LIQUID HYDROGEN FLEXIBLE DUCTING TECHNOLOGY

SUBMITTED TO

National Aeronautics and Space Administration  
George C. Marshall Space Flight Center  
Huntsville, Alabama

PREPARED BY :

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Industrial Products



ER 1473- 4

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## INTRODUCTION

This is the fourth progress report <sup>(first quarterly)</sup> on the study of liquid hydrogen flexible ducting technology being performed under the National Aeronautics and Space Administration Marshall Space Flight Center Contract Number NAS8-1103. This report is also the first quarterly progress report. The period covered by this report is 15 July through 31 October 1964.

This program is being conducted under the direction of Solar Aerospace Engineering with Mr. H. T. Mischel as Program Manager.

During the previous reporting period, a material survey performed on a program presently in progress with the Marshall Space Flight Center Manufacturing Engineering Laboratory had been completed and the data was included as an appendix of that report. The previous report also included a flange concept utilizing solder as the strength and sealing member and this concept was expanded during this reporting period. A study on the optimization of gimbal joints was initiated and was completed during this reporting period.

### Gimbal Joint Optimization

A study to optimize gimbal joint designs was completed during this reporting period and is included as Appendix A.

The major goal of this study was weight reduction with increased reliability, increased flexibility and reduced pressure instability.

The study was based upon a gimbal joint that Solar is presently fabricating. This gimbal joint is used on the S-II vehicle in the hydrogen tank pressurization system.

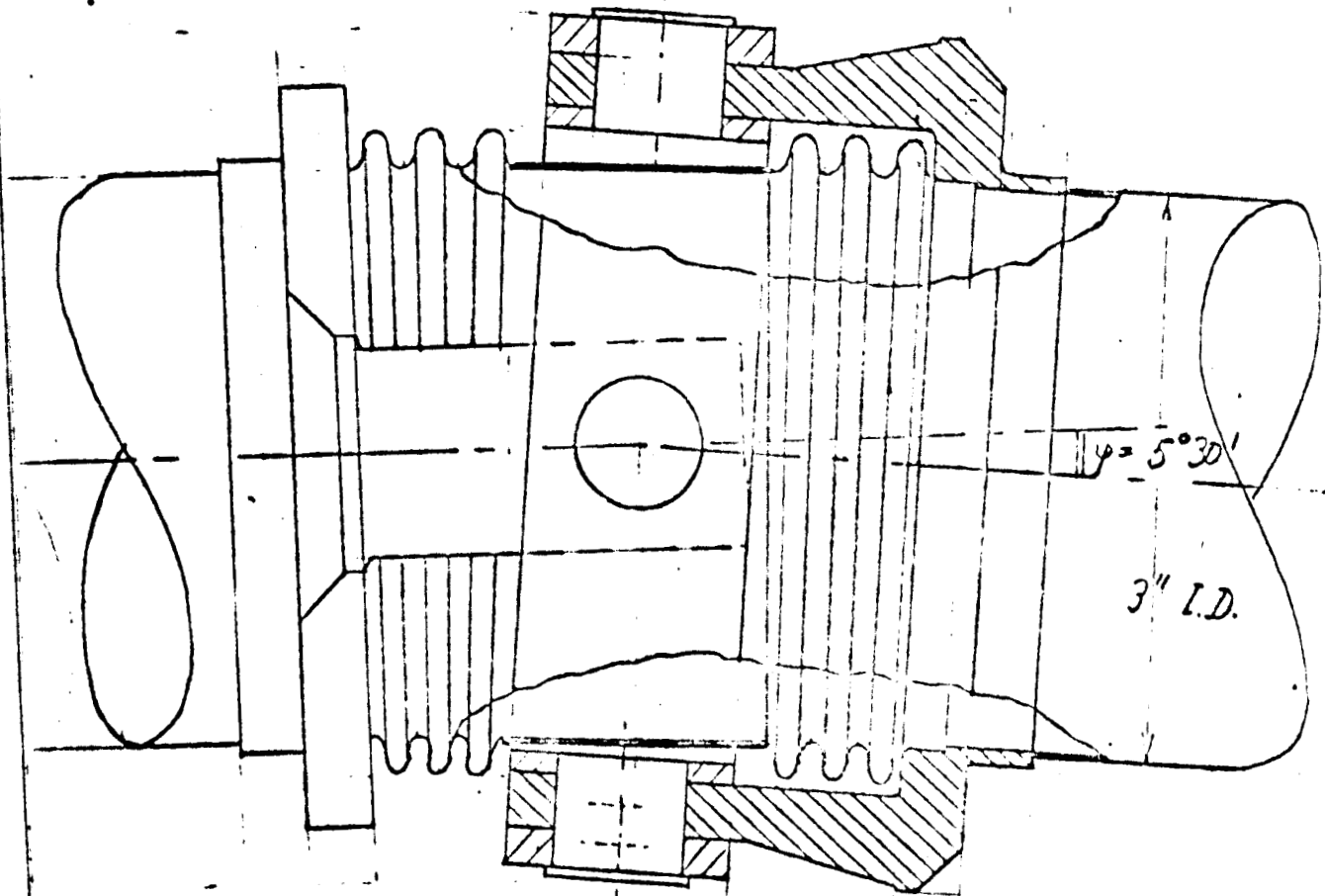
The study was made in three parts: (a) the gimbal joint was theoretically treated and the results applied in developing a light weight gimbal joint design; (b) a study was made to reduce the weight of a given joint by simple material substitution; and (c) the gimbal joint is compared to several other possible joint concepts. An interesting idea which has been developed in this study is the eccentric application of the pin load to the gimbal rings resulting in a torsional moment which is opposite to the torsion developed in the ring at 45-degrees to the pin locations. The study shows a curve for the effect of this eccentricity on the torsional shear stress field in the ring. Various methods of applying this torque are discussed and stress analysis for these methods are shown.

As a result of this study, some interesting concepts have been developed which it is felt should be the subject of future work. One of these concepts is the placement of the gimbal ring at approximately the same diameter as the bellows (see Figure 1) to reduce the line weight by reducing its size and also its applied moment. This would necessitate dividing the

## Reduced-dia-gimbal-ring

Reduction of ring-dia: old O.D.: 5" } reduction 10%  
new O.D.: 4.5"

The lugs must be stronger because of the greater bending-moment resulting from the across-load.



The flanges may remain the same: they have to take the bigger bending-moments from the across-load, but the basic strain is smaller (lugs closer to center). The squirm-behaviour of the bellows isn't improved. Light-weight hollow profile cannot be used.

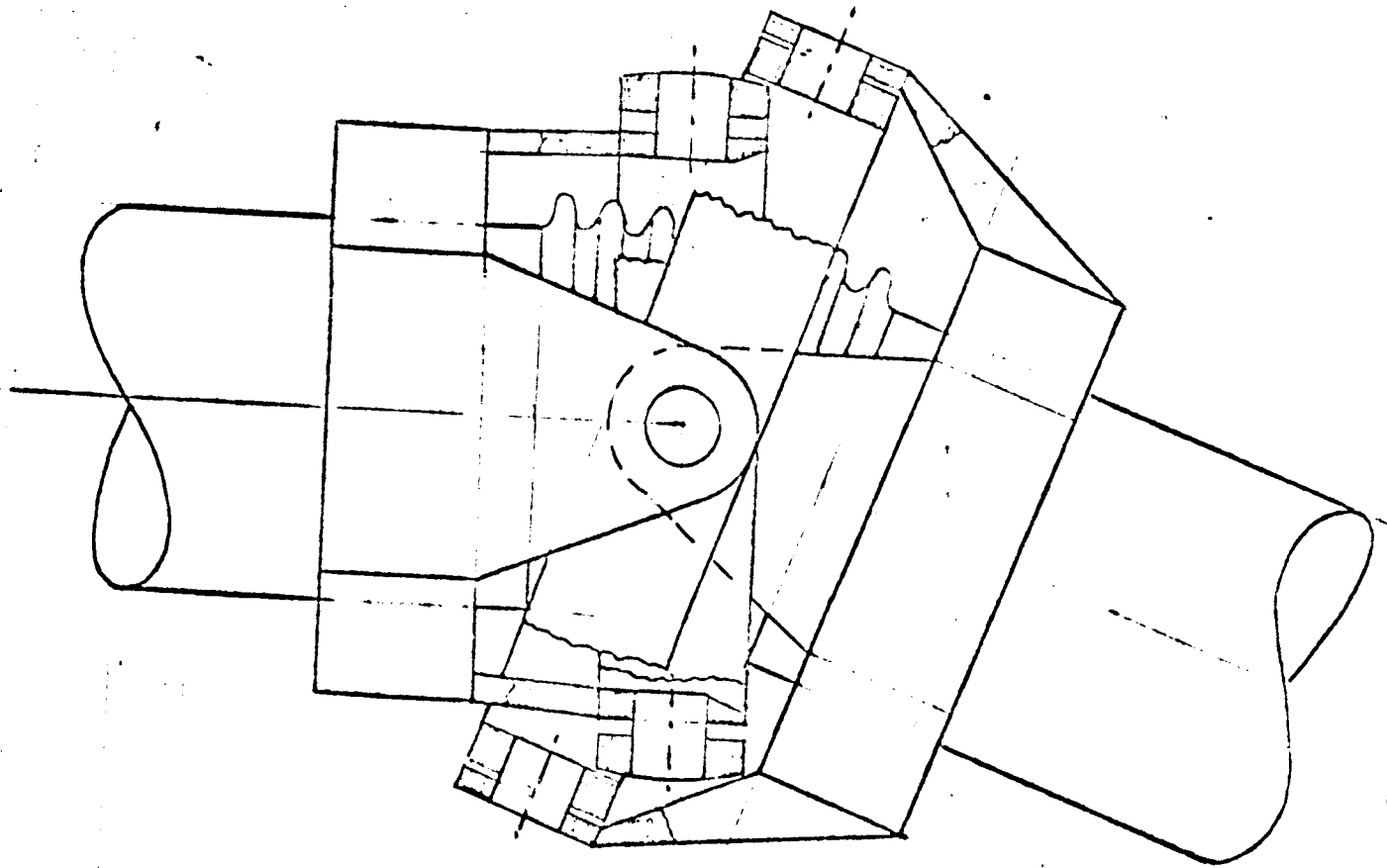
FIGURE 1

bellows into two separate elements. As a result a study of this particular gimbal joint concept will be undertaken in the next reporting period.

Another interesting concept is an eight lug gimbal joint (shown in Figure 2). The purpose of the four lugs on each flange is to apply the pin load to the flange at more than 2 points as is presently done. The advantages of this approach are not too readily seen on the small diameter units, the 3-inch size being the subject of study of this program, but shows itself to be more advantageous in the larger diameters where 2 point loadings induce extremely high moments in the lug flange. Compounding the problem, the thin wall duct adjacent to the gimbal is generally incapable of assisting in the resistance of this moment and must receive this load as uniformly distributed tension. The study of the gimbal joints reveals the necessity for further work with regard to the gimbal rings and a study to investigate the effect of the eccentricity and the equations which determine the torsional shear stress has been initiated and will be completed during the next reporting period.

## Double-ring-joint

An obvious disadvantage of the gimbal-joint is the load-concentration to two points. The following solution enables a load distribution to four points instead of two:



Advantage: Load-distribution to 4 points on tubes.

Disadvantages: Complicated, big outside dia, rings on tubes loaded with very high neg. eccentricity.

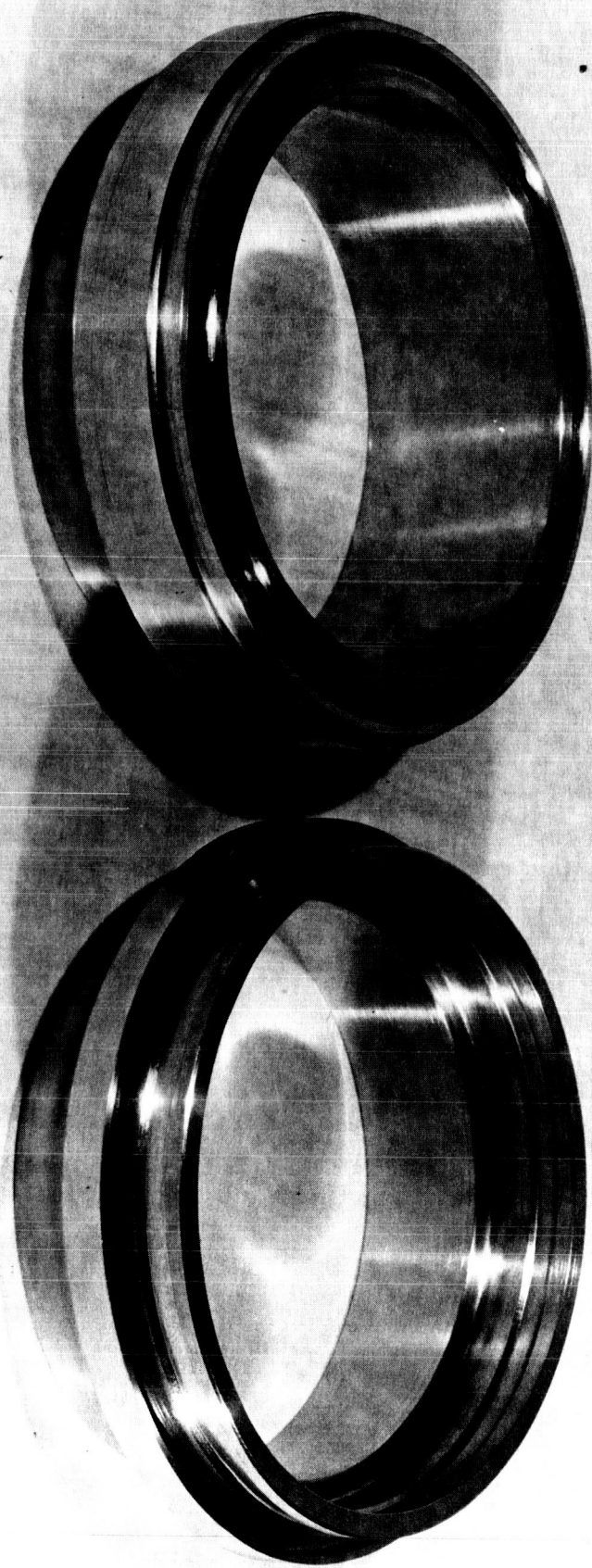
FIGURE 2

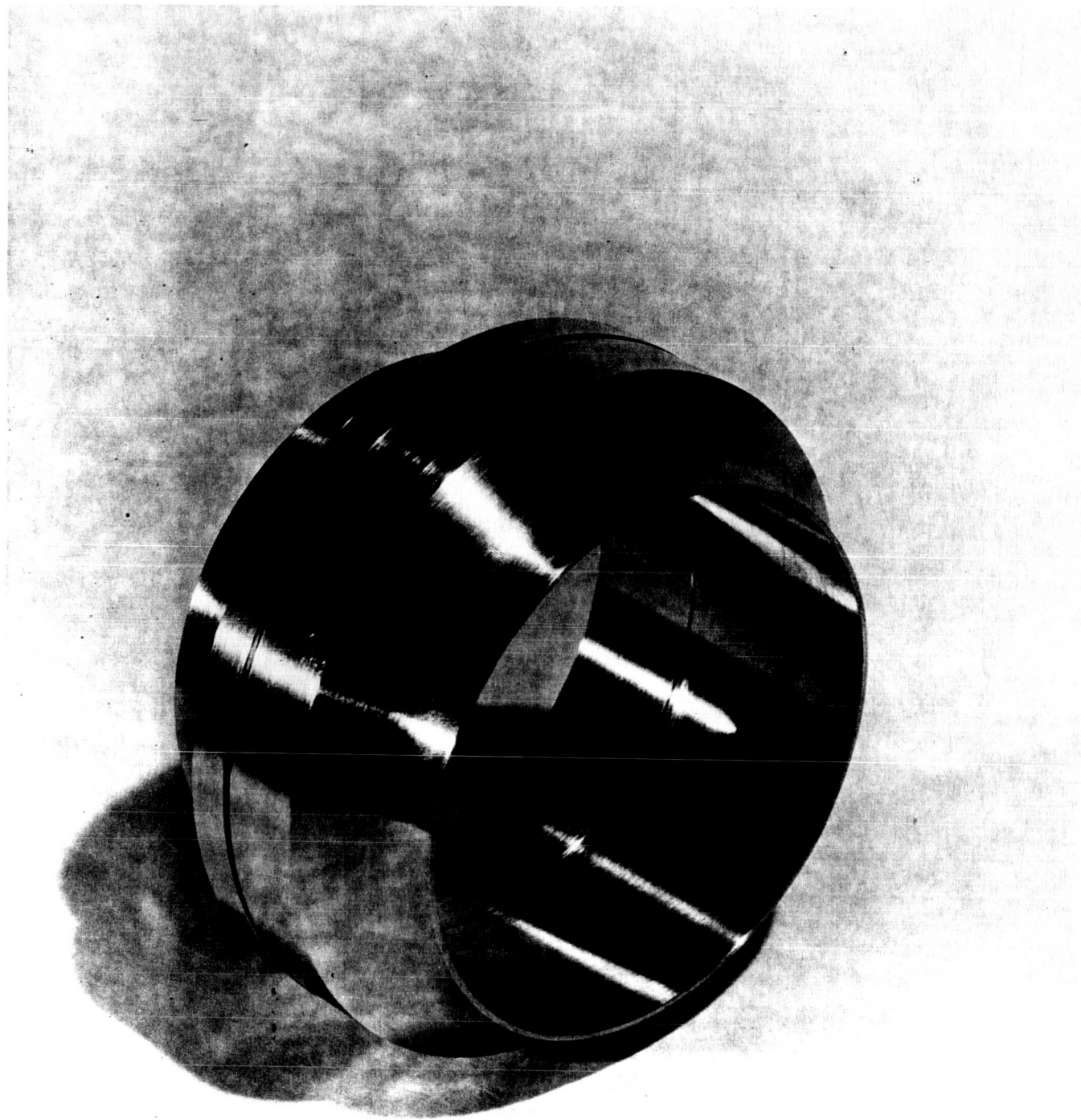
### The Boltless Flange Concept

One each of the male and female boltless flange have been fabricated during this reporting period. Photographs of the assembly are shown in Figure 3. The boltless flange was welded to tube segments with end caps and the joint was made with Claude-Michael #20 solder. This joint was subjected to a pneumatic pressure test under water and was found to leak. The joint was remade by placing it in an oven until the solder was molten and replacing the solder in the groove for a second attempt. The joint again was found to leak. The joint was taken apart by the method previously described, a solder flux was applied and the joint reconnected. This was found not to be much help. It is interesting to note, however, that no solder was observed on the inside of the connection during these processes. Discussions with Solar metallurgical engineers indicate that the use of the solder has two disadvantages: the solder contracts when cooled and breaks away from the walls of the joint and the solder with its inherent incompatibility or non-wetting characteristics when in contact with stainless, tends to ball up in the joint and therefore permit leak passages.

The joint was therefore cleaned of all solder and was remade with Cerabond, an alloy of the Cerro-DePasco Company. This alloy is commonly used in forming operations. While its melting point is at approximately 190°F, the alloy does exhibit good properties at cryogenic temperatures and is currently being used by NASA-Lewis as was described in the original NASA data sheet.

The joint was subjected again to a pneumatic pressure test under water and it exhibited no leaks in the low pressure range. The pressure was increased and the joint exhibited leaks at approximately 50 psi. The test was





not terminated at this point, however, in order to demonstrate whether the joint had sufficient strength to withstand the full pressure end load. The pressure was increased to 150 psi and no change in the leakage rate or distortion in the joint was observed. The joint was then reheated, the seal broken and the joint remade. It is intended that the joint be subject to a mass spectrometer leak check in the next period to determine whether the joint in the low pressure range is exhibiting vacuum tightness.

### Non-Vacuum Jacketed Insulation

The insulation that appears to be most promising is a TFE (teflon) wool which is currently produced by Shamban Associates, which would be wrapped around a duct with an adequate radiation shield material such as aluminum foil. This buildup would be covered by a heat shrinkable teflon tube and filled with CO<sub>2</sub> gas at ambient pressure. When the duct is flowed with liquid hydrogen the insulation would be evacuated by cryopumping the CO<sub>2</sub> tending to compress the insulation around the duct. While this is not a particularly good insulation in the atmosphere, the theory is that the teflon wool would have sufficient resiliency at -423°F to re-expand the teflon outer tube in the vacuum of space and therefore provide an excellent insulation for space missions such as parking orbits and interplanetary travel. Since liquid hydrogen and liquid oxygen systems are predominantly used for upper stage use, the assumption is that the insulation is not a particular requirement in atmosphere, and would therefore provide its most efficient mode when the duct is flowed in the vacuum space. Contact will be made with Shamban Associates to obtain prices and delivery schedules for sufficient amount of the teflon wool for an experiment to prove this concept. The experiment itself will be performed on the liquid hydrogen duct experimental device which is currently being designed for the vacuum jacket study contract no. NAS8-11340.

## APPENDIX A

### GIMBAL JOINT OPTIMIZATION

SUBJECT: NASA-Gimbal-joint-  
optimisationBY: D. Hegg

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## NASA-Gimbal-joint-optimisation

In aerospace-problems weight-reduction is a major goal. An attempt will therefore be made to reduce the weight of the gimbal-joint of Dwg. 38770. This work consists of three parts:

1. In a first part a gimbal-ring will be theoretically treated and the results applied in constructing a light-weight gimbal-joint without changing the basic concept.

In a second part a study is made to reduce the weight of the given joint (Dwg. 38770) by simple material-substitution.

In a third part the gimbal-joint is compared to several other possible joints (existing ones and new ideas). The question must be raised whether an other solution would be advantageous.

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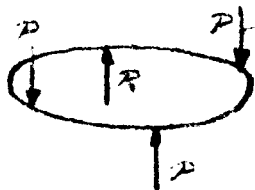
SUBJECT: NASA-Gimbal-joint-  
optimisationDATE 10-28-64BY: O. AbeggPAGE 1 OF \_\_\_\_\_ PAGES

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In the given gimbal-assy both, gimbal-ring and gimbal-lug should be considered to try to reduce their weight. An attempt will therefore be made to optimise those parts.

### Theoretical treatment of gimbal-ring:

The most simple case is that of the ring equally loaded by 4 forces in axial direction:

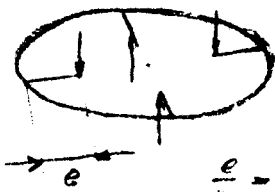


pin-force  $P = 1/2$  separating force  $F$

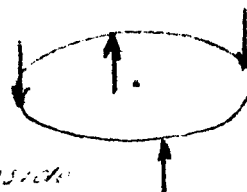
Max bending moment:  $\frac{Pr}{2}$  in plane of pins

Max torsional moment:  $\frac{Pr}{2} (0.414)$  at  $45^\circ$  between pins.

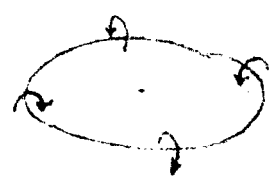
If the ring is eccentrically loaded there occurs an additional moment at each pin:



is equal to super-  
position of:



and



$\frac{e}{r} = \epsilon$   $\epsilon > 0$  if load inside  
 $\epsilon < 0$  if load outside

The first case is equivalent to the case above. The second case can be analysed and gives the following results:

Bending moment:  $-\frac{Pr}{2} \epsilon$  at pins

Torsional moment:  $\frac{Pr}{2} \epsilon$  at pins  
 $-\frac{Pr}{2} \epsilon \sqrt{2}$  at  $45^\circ$

The moments at  $30^\circ$  and  $60^\circ$  can easily be analysed in the same way. The results of the two cases must now be superimposed:

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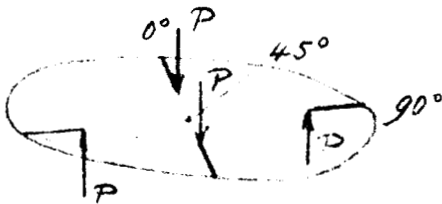
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Superposition:

at pin

TorsionBend $30^\circ$ 

$$-\frac{Pr}{2} \epsilon$$

$$\frac{Pr}{2} (1-\epsilon)$$

 $45^\circ$ 

$$\frac{Pr}{2} (0.366 - 1.366 \epsilon)$$

$$\frac{Pr}{2} (0.366 - 0.366 \epsilon)$$

 $60^\circ$ 

$$\frac{Pr}{2} (0.414 - 1.414 \epsilon)$$

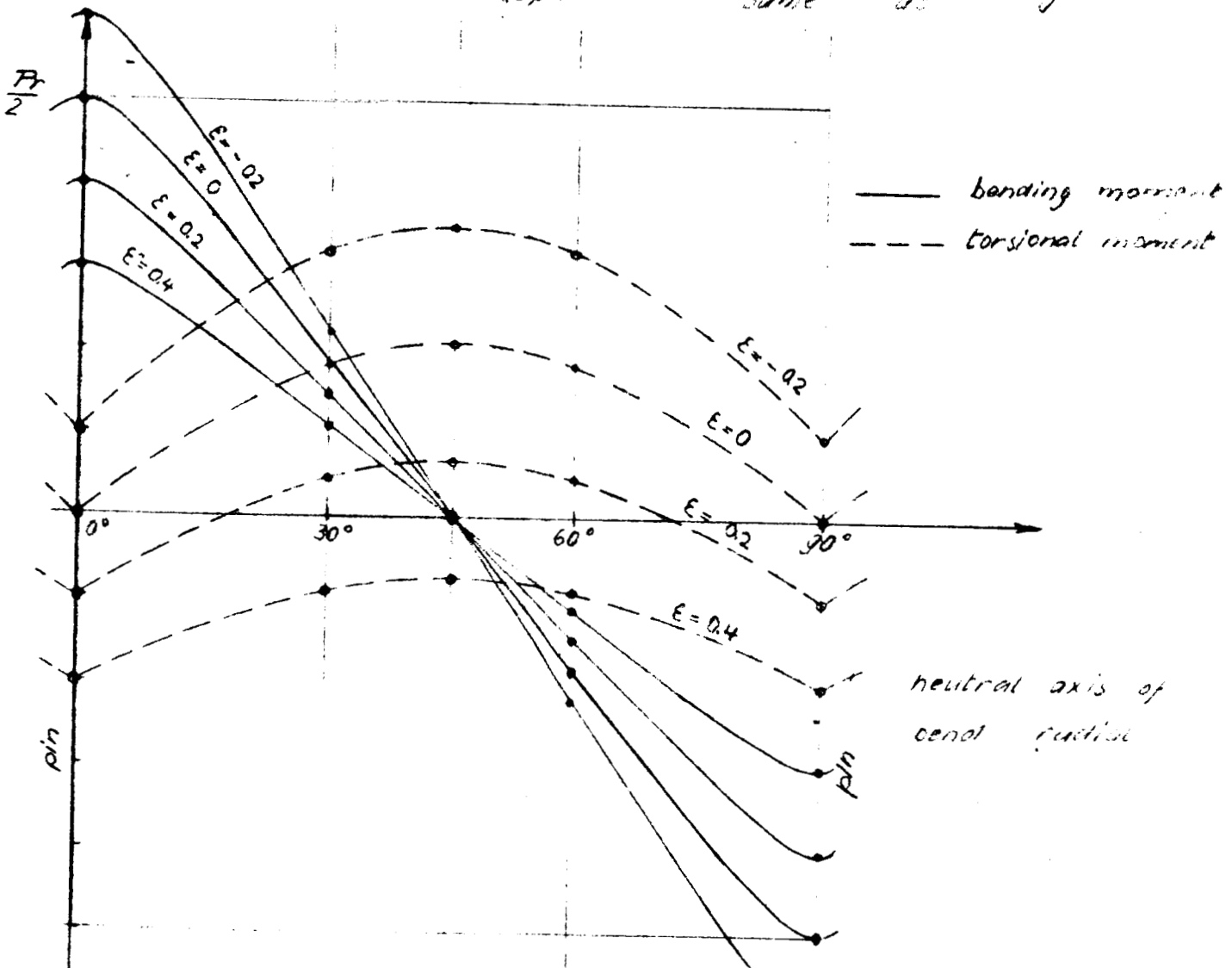
 $30^\circ$ 

same as

 $90^\circ$ 

at pin

same as



Conclusions: It is very dangerous to apply the forces outside the ring, but it is favorable to have a small positive  $\epsilon$ .

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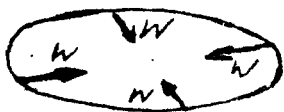
SUBJECT: NASA - Gimbal - joint  
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 BY: J. Hegg

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In a more general case the applied forces may have a radial component; this case can be found in Roark p. 156:

No. of loads: 4,  $\theta = 45^\circ$

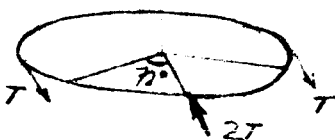


Bending moment:  $M = Wr(0.137)$  at pins  
 $M = Wr(0.071)$  at  $45^\circ$   
 Neutral axis of bend: axial.

If the gimbal-joint has an angular deflection, there occurs an across load:

It will be showed later

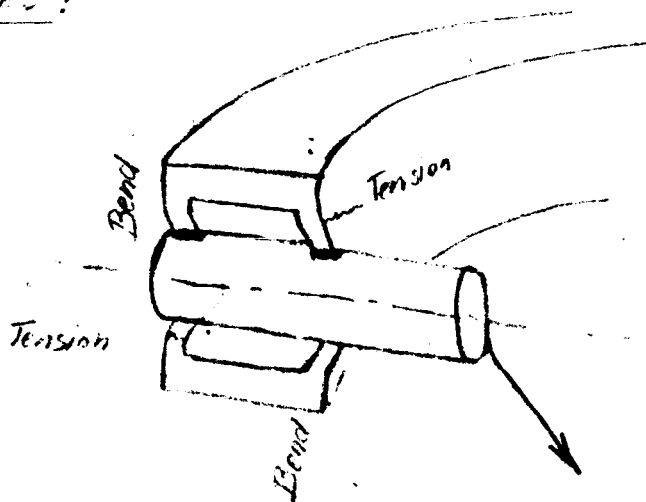
why this case occurs  
 and not (see p. 10)



This case is found in Roark p. 157:

Bending moment  $M_{max} = Tr \begin{cases} 0.5 & \text{at pins} \\ 0.407 & \text{at } 45^\circ \end{cases}$   
 Neutral axis of bend: axial

Where the pins are welded to the ring there are additional stresses:



Tension: stress =  $\frac{\text{load}}{\text{area}}$

Bend:  $M_{max} = \frac{(T \cdot h)}{8}$  at load  
 Diagram shows a vertical line with a horizontal arrow pointing right labeled  $\frac{T \cdot h}{2}$ .

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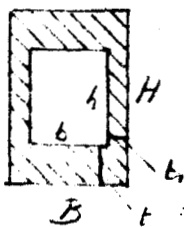
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From the moments analysed in page 1, 2 and 3 the stresses can easily be computed for every part of the ring:

Tensile stresses:  $\sigma_{\text{ten}} = \frac{M_{\text{bend}}}{W_{\text{bend}}}$

Shear stresses:  $\tau_{\text{max}} = \frac{M_{\text{torsion}}}{W_{\text{torsion}}}$

I will choose a hollow box beam profile for the ring which is equally suitable for bend and for torsion:



$$W_{\text{bend}} = \frac{I}{c} = \frac{BH^3 - bh^3}{6H}$$

$$W_{\text{torsion}} = \frac{I}{c} = \frac{(B+1.4h)t}{2}$$

$$W_{\text{torsion}} = (H-t)(B-t)2t \quad \text{near mid-length of short sides}$$

$$W_{\text{torsion}} = (Ht)(B-t)2t \quad \text{near mid-length of long sides}$$

(see "Steel I",  
Springer-Verlag, Berlin)

(Roark)

As soon as all stresses are calculated the max equivalent stress must be found (v. Mises):

$$\sigma_0 = \sqrt{\sigma_x^2 + \sigma_y^2 + \sigma_z^2 - \sigma_x \sigma_y - \sigma_x \sigma_z - \sigma_y \sigma_z + 3\tau_{xy}^2 + 3\tau_{xz}^2 + 3\tau_{yz}^2}$$

Since the ring must not be welded to the tube, I am free to choose the material:

alloy titanium 5 AL - 2.5 Sn

Tensile strength: 186,000 psi	} at -320°F	118,000 psi	} at RT.
Yield strength: 184,000 psi		113,000 psi	

(progress report 2)

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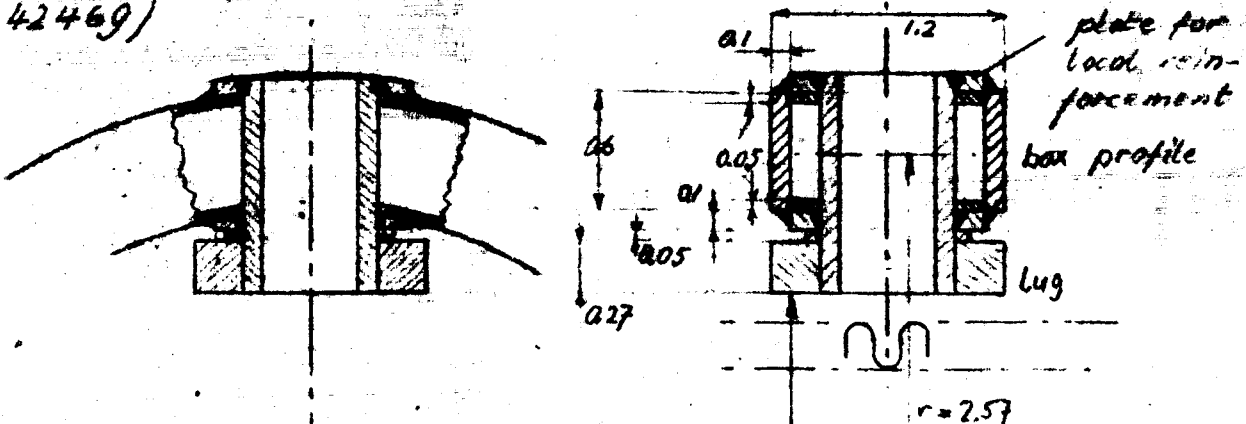
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Construction of gimbal - ring (Dwg. 42469)

To get a gimbal ring with optimum performance the following principles should be watched:

- Load ring eccentrically :  $0 < \epsilon < 0.2$  (follows from p. 2)
- Choose a hollow profile that is equally adequate for bend and torsion: The hollow box beam profile is better than the circular ring-profile because: Better performance against bending; smaller  $\epsilon$  possible
- Avoid notches to allow to keep margin of safety small.
- Prevent uncontrolled influences such as bend from lug.

These considerations lead to the following construction:  
(see Dwg. 42469)



Dia  $\left. \begin{matrix} 3.68 \\ 3.69 \end{matrix} \right\}$  compare to Dwg. 38770

The profile is locally reinforced to keep the additional stresses (p. 3) low. The original profile (Dwg 38770) has approx same outside-profile but it is much heavier because it is not hollow and because it is made out of steel.

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Computation of ring-stresses

Operating data (from Dwg. 38770)

Pressure: Operating 860 psig no pressure-cycles  
 Proof 1290 psig considered (static load)  
 Burst 2150 psig

Temp. :  $-320^{\circ}\text{F}$  It is assumed that all parts  
 take this temperature.

Yield- and tensile strength are almost same for titanium. The stresses will therefore be computed for burst-pressure at which the yield-strength may be reached. However a margin of safety must be allowed because:

Notches (pin-hole, weld) include local high stresses  
 Higher temperatures might occur temporarily at some locs.  
 Irregularities in material

$$\text{Separating force } ^1 F = 2150 \cdot \pi \cdot 1.58^2 \quad (R_m = 1.58)$$

$$= 16880 \text{ lbs} \quad (\text{burst})$$

$$\text{Pin force } P = 8440 \text{ lbs} \quad (\text{burst})$$

$$\text{Radius } r = 2.57" \quad (\text{see p. 5, center of profile})$$

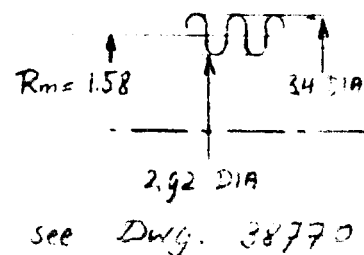
$$\text{Eccentricity } E = 0.21 \quad (\text{see p. 11})$$

$$\text{Profile } \text{see p. 5}$$

$$\text{Radial compon. } W = 1190 \text{ lbs} \quad (\text{see p. 11})$$

$$\text{Across load } T = 415 \text{ lbs} \quad (\text{see p. 11})$$

No torque



<sup>1)</sup> In the separating force no bellows-processing included.

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Axial load  
 (see p. 2, 4)

Stresses at pins

Tension:

$$M = \frac{8440 \cdot 2.7 \cdot 0.21}{2} = 2280 \text{ lbs}$$

$$W = (0.72 \cdot 0.5) \cdot 0.05 = 0.061$$

$$W = (0.72 \cdot 0.5) \cdot 0.10 = 0.122$$

$$\tau = \frac{2280}{0.061} = 37400 \text{ (long side)}$$

$$\tau = \frac{2280}{0.122} = 18700 \text{ (short side)}$$

Bend:

$$M = \frac{8440 \cdot 4.5 \cdot 0.19}{2} = 3500 \text{ lbs}$$

$$W = \frac{0.617 \cdot 0.51}{6.12} = 0.0378$$

$$\sigma = \frac{3500}{0.0378} = 97500 \text{ (short side)}$$

$$M = \frac{1190 \cdot 2.7 \cdot 0.17}{2} = 419$$

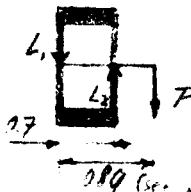
$$W = \frac{12.22 \cdot 0.5 \cdot 0.15}{6.06} = 0.0372$$

$$\sigma = \frac{419}{0.0372} = 11300 \text{ (long side)}$$

$$M = \frac{405 \cdot 2.57 \cdot 0.15}{2} = 520$$

$$W = 0.0372$$

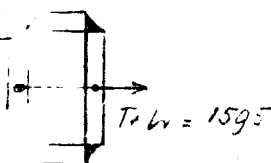
$$\sigma = \frac{520}{0.0372} = 14000 \text{ (long side)}$$

Tension Load

$$L_1 = \frac{8440 \cdot 0.14}{0.7} = 2290 \text{ lbs}$$

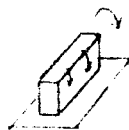
$$L_2 = \frac{4040 \cdot 0.14}{0.7} = 10750 \text{ lbs}$$

$$\sigma = \frac{211.2 \cdot 0.2}{0.12} = 54000 \text{ psi}$$

Bend:

$$L_2 = 0.4$$

$$M = \frac{1595 \cdot 0.4}{16} = 40$$



$$W = \frac{0.617 \cdot 0.5}{6} = 0.00167$$

$$\sigma = \frac{40}{0.00167} = 24000 \text{ psi}$$

Radial compression  
 (see p. 5)  
 Bend only

Uniaxial stress  
 (see p. 3)  
 Bend only

Radial compression  
 (see p. 3)

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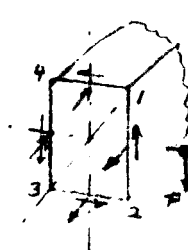
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Max equivalent stress: (see p. 4)

The extension is made for the plane of the pin. At 45° see p. 4.

a) Without additional stresses:





Middle of pin side:  $\sigma_e = \sqrt{97800^2 + 348700^2} = \sqrt{140110000} = 109000$

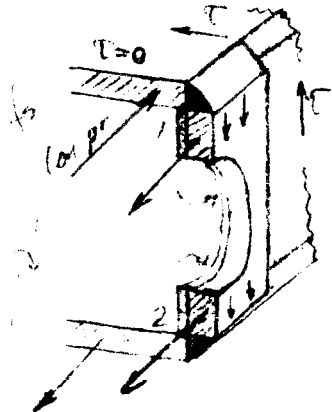
Middle of leg side:  $\sigma_e = \sqrt{\frac{43200 + 43200}{25300} + 3 \cdot 37400^2} = 41100$

Corner:  $\sigma_e = \sqrt{(97800 + 25300)^2 + 3 \cdot 37400^2} = 139000$

Safety factor at corners 2 & 4:  $n = \frac{184000}{139000} = 1.32$

b) Additional stresses:

On p. 7 the tensile stress is calculated as if:  but actually: . Thus " " actually smaller. Because the pin does not have an infinite cross size, the stress vanishes in the mid-plane.



Corner 2: (outer edge)

$$\sigma_e = 1000 \sqrt{(978 + 253)^2 + (54 + 24)^2 + (978 + 253)(54 + 24)} = 176000 = \text{max equivalent stress}$$

Corner 1: (inner corner)

$$\sigma_e = 1000 \sqrt{(978 - 253)^2 + (54 + 24)^2 + (978 - 253)(54 + 24)} = 131000$$

Stress in corner 3 & 4 for insertion ( $L_1 \ll L_2$ , see p. 7)

Safety factor of corner 2:  $n = \frac{184000}{176000} = 1.05$  (at burst)

The value of 94500 is too low, however, as shown above. The gimbal ring will not burst at burst pressure. However local yield might occur in welds.

Important: Based on operating pressure: Safety factor =  $\frac{105 \cdot 2150}{100} = 2.262$

If the NMA - margin (10% for yield, 40% for UTS) is included, this factor reduces to  $(2.262) \cdot 1.87$  (1.4 because yield is 47%), and if considering factor = 1.4 prestressing (503 lbs, see M-1697 P. 32) it reduces to  $1.87 / 1.075 = 1.74$ . Thus:

Margin of safety to NMA - standards =  $1.74 - 1 = 0.74$

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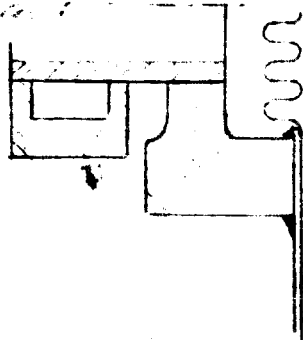
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SUBJECT: NASA Gyrobar, 245-DATE 10-28-64OptimizationPAGE 1 OF \_\_\_\_\_ PAGESBY: W. H. H. H.

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Construction of Gyrobar, assembly

The problem is to distribute the load from the gimbal-ring and the pins to the tube as equally as possible. The following approach is being attempted:



Load partly transmitted to the tube  
on direct way:

uncontrolled stress peak in tube:

Load partly distributed:

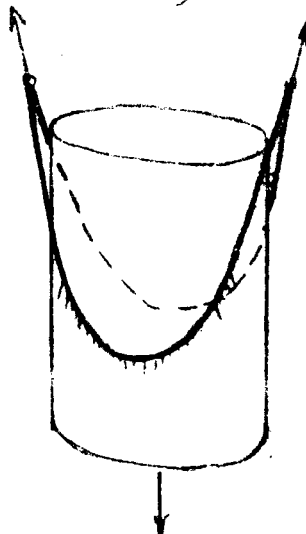
(compare to page 3;  $\epsilon < 0$ )

unfortunate stress peaks at pins:

The profile is full cast very heavy.  
Through deformation an uncontrolled  
moment is transmitted to the  
gimbal-ring:

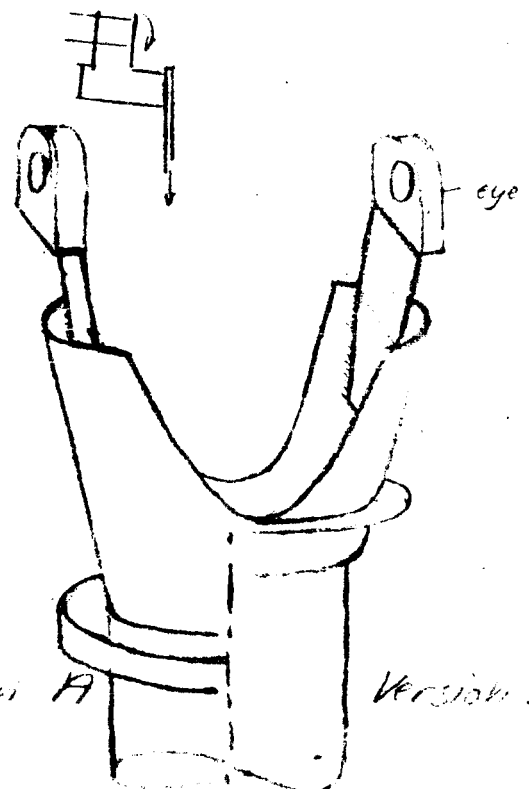
The following solution tries to avoid  
those disadvantages:

Idea:



Design:

Version A



Version 2

## ENGINEERING REPORT

SOLAR AIRCRAFT COMPANY

NO. \_\_\_\_\_

SUBJECT: NAS-1 - Gimbal - joint  
optimizationDATE 10-28-64PAGE 16 OF \_\_\_\_\_ PAGESBY: W. H. G. J.

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The idea is to eliminate the load by pure tension. thereby, since there is no internal joint and cone is uniformly material. The eye is connected to the cone with a flexible member, thus avoiding concentrated bending moments.

One fact is obviously unfortunate: The stress-distribution in the cone is completely unknown. To avoid stress peaks where the cone is welded to the eye there can be either a

reinforcement - ring (Version A)

or the cone will be allowed to fold (Version B)

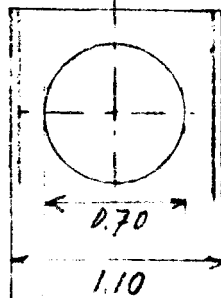
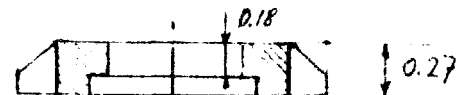
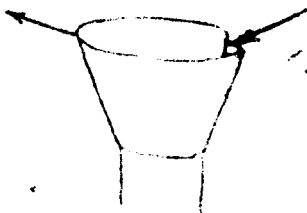
The better version must probably be found by test.

Eye The eye from Dwg 38770 must be changed because the cone cannot transfer loads inward but only outward. (see also p. 3).

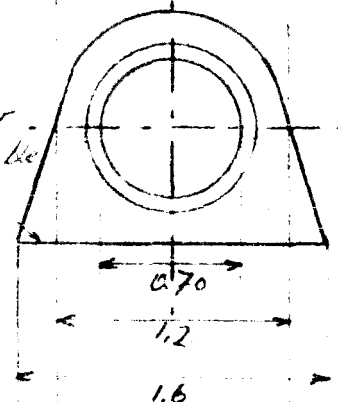
Radial components and across loads are therefore transmitted by tension in the pin

old eye:

new eye:



long edge for  
weld to flexible  
member



New eye has lower stresses than old one and will therefore not be compressed.

New bearing pressure:  $p = \frac{2440}{0.7 \times 1.6} = 67000 \text{ psi}$   
(26800 psi at opening)

A proper MoS<sub>2</sub> - dry lubrication must be provided.

Bearings with a pressure up to 28000 psi are in operation.

## SOLAR AIRCRAFT COMPANY

## ENGINEERING REPORT

SUBJECT: NASA - Gimbal - joint -  
flexible member  
 BY: C. H. G.

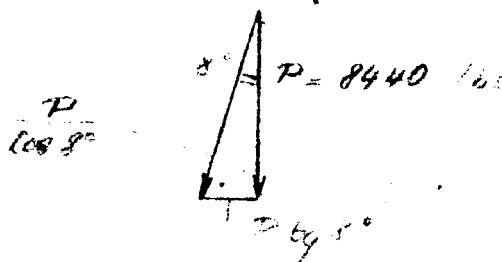
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The eccentricity  $e$  can now be found by checking the distance from the bearing center to the center of the profile.

$$e = 0.54 \quad e = \frac{0.54}{2.57} = 0.21 \quad (\text{see p. 6})$$

Flexible member: Cross-section at place where welds to come,  
 $1.4 \cdot 0.05 = 2.07 \text{ in}^2$

Load: The angle of  $8^\circ$  is chosen:



The load follows from the figure = 8530 lbs

Raxial component (see p. 6)  
 $W = 1190 \text{ lbs}$

Tensile stress:  $\sigma = \frac{8530}{2.07} = 122'000 \text{ psi}$  Internal HB

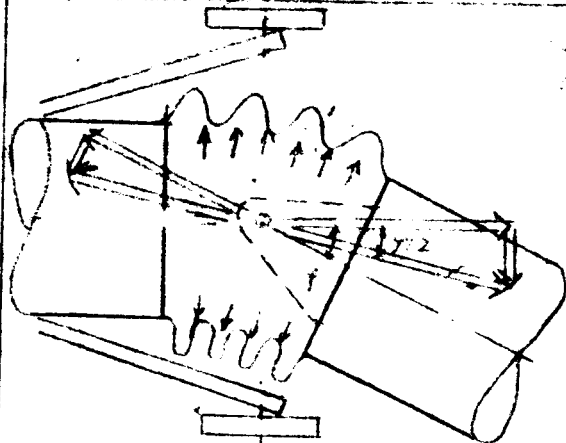
Pin: Bending moment:  $M = 8440(0.84 - 0.7) = 1605 \text{ lb-in}$   
 $W = \frac{\pi}{32} \frac{D^4 - d^4}{D} = \frac{\pi}{32} \frac{0.24^4 - 0.20^4}{0.7} = 0.052$  }  $\sigma = \frac{1605}{0.052} = \pm 64'200 \text{ psi}$

Shear: Load = 8440 lbs  
 Area:  $\frac{\pi}{4} (D^2 - d^2) = \frac{\pi}{4} (0.24^2 - 0.20^2) = 0.188$  }  $\tau = \frac{8440}{0.188} = 44'800 \text{ psi}$

Tensile stress:  $(2T + W) = 2100 \quad \sigma = 64'200 \cdot 2100/8440 = 16'000 \text{ psi}$

Equivalent stress:  $\sigma_e = \sqrt{64'200^2 + 16'000^2 + 64'200 \cdot 16'000 + 2 \cdot 44'800^2} = 106'000 \text{ psi}$

Angular deflection of joint:



It can easily be seen that an areas load occurs when the joint is deflected:

$$2T = 16880 \sin 5^\circ = 810 \text{ lbs} \quad y = 5^\circ \text{ in } 1 \text{ see p. 3, 6}$$

This areas load must be transmitted from the eye to the base. The flexible member cannot take it.

Wdg. 42469 must therefore be completed. Further alterations will be made too.

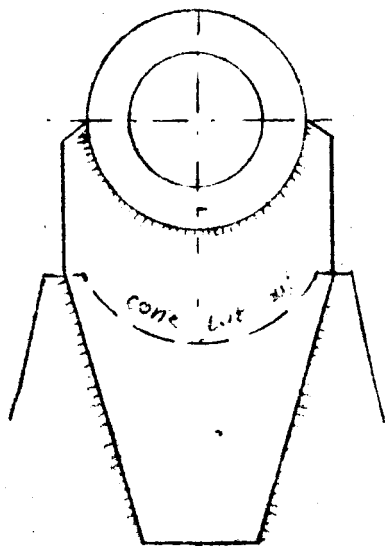
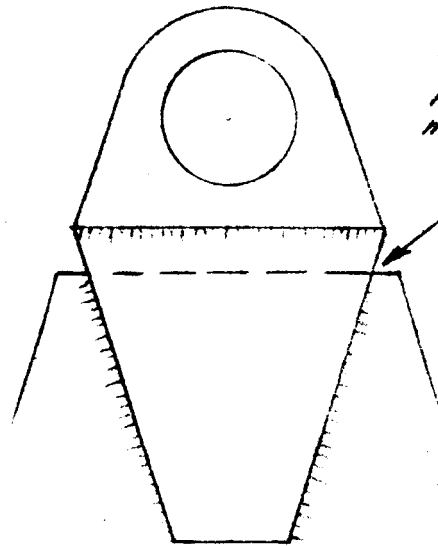
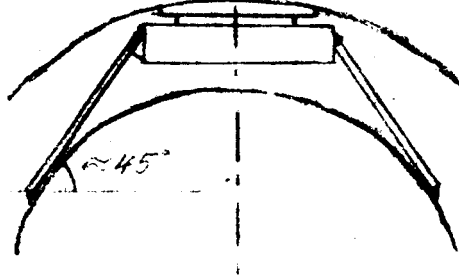
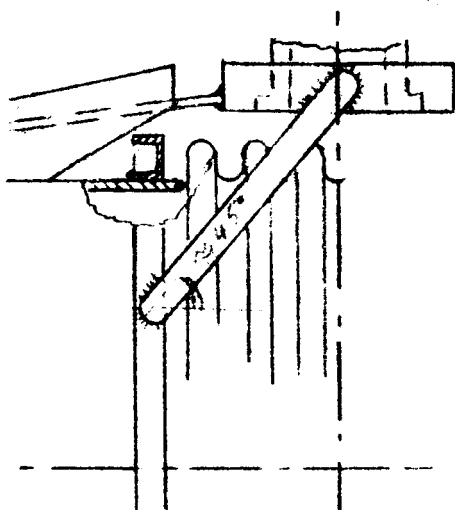
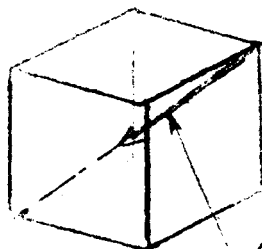
## SOLAR AIRCRAFT COMPANY

## ENGINEERING REPORT

NO. \_\_\_\_\_

SUBJECT: NASA - Gimbal joint -DATE 10-28-64optimizationPAGE 12 OF \_\_\_\_\_ PAGESBY: J. Hagg

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Iteration: (Dwg. 42469):New eye with new flexible member:Old eye:In order to take the across-load an additional element will be provided:Across load:  $2T = 810 \text{ lbs}$ 

$$\frac{810}{2} = 405 \text{ lbs}$$

$$\sqrt{3} \cdot 405 = 700 \text{ lbs}$$

Necessary area  
 $\sigma = 70000$   
 (assumption)

$$\text{area} = \frac{700}{70000}$$

$$= 0.01 \text{ in}^2$$



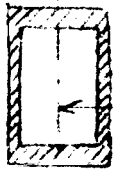
SUBJECT: NASA-Gimbal-joint -  
optimization  
 BY: D. Hegg

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## Discussion:

The gimbal-joint of Dwg 38770 is very heavy. An attempt has therefore been made to reduce the weight without changing the basic concept.

A good solution has been found for the gimbal-ring.  
Weight of new ring:



$$F = 0.612 - 0.51 = 0.102 \quad (\text{see p. 5})$$

$$r = 2.57$$

$$\text{volume} = 2\pi r F$$

$$= 3.55 \text{ in.}^3$$

$$8 \text{ reinforcement plates: } 8 \cdot 0.111 = 0.88$$

$$\begin{array}{r} \text{Total} : 3.55 \\ + 0.88 \\ \hline 4.43 \end{array}$$

$$\text{Specific gravity of titanium: } 0.163 \frac{\text{lb}}{\text{in.}^3}$$

$$\text{Weight of ring: } 4.43 \cdot 0.163 = 0.72 \text{ lb.}$$

Weight of old ring: (Dwg. 38770)



$$F = 4.28 - 3.68 = 0.60$$

$$r = 2.14$$

$$\text{Vol} = 2\pi r F$$

$$= 9.52 \text{ in.}^3$$

$$12 \text{ holes } 0.74 \text{ dia: } 12 \cdot \frac{\pi}{4} \cdot 0.74^2 \cdot 0.6 = 3.1$$

$$\begin{array}{r} \text{Total: } 9.52 \\ - 3.10 \\ \hline 6.42 \end{array}$$

$$\text{Specific gravity of steel: } 0.282 \frac{\text{lb}}{\text{in.}^3}$$

$$\text{Weight of ring: } 6.42 \cdot 0.282 = 1.81 \text{ lb.}$$



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SUBJECT: NASA - Gimbal-joint -  
optimisation

BY: O. Heagy

Reduction in weight:

Old ring: 1.81 lb

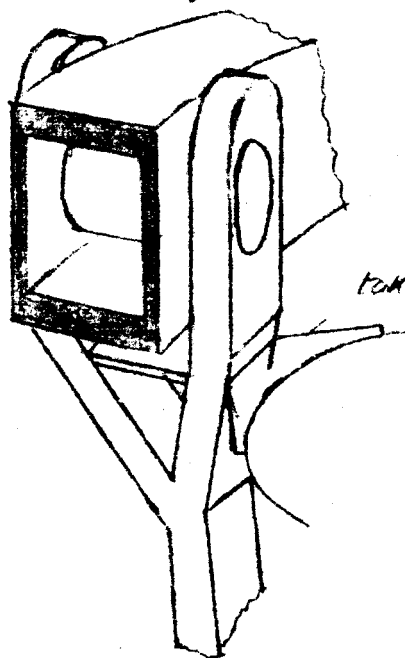
New ring: 0.71 lb

Reduction: 1.1 lb  $\approx 60\%$

The weight of the new ring is reduced to about 40% of what the old ring weighed.

Further possibility of weight-reduction:

The eccentricity  $\epsilon$  (see p. 1, 2, 11) has been made as small as possible;  $\epsilon = 0.21$ . This value is above the optimum. The following solution allows any arbitrary  $\epsilon$  and has other advantages too:



take across-loads

If the eccentricity is made so that the force acts on the inner edge of the ring, the additional bending moment (see p. 3.7) can be completely avoided. The additional tensile stress reduces to  $\frac{8440}{54000} \text{ psi} = 42.400 \text{ psi}$ . The original bending stress 10750 in ring increases to  $97800 \cdot \frac{0.865}{0.79} (\text{new } \epsilon = 0.135) = 107000$ .

The max equivalent stress (see p. 8) reduces to 158500 psi.



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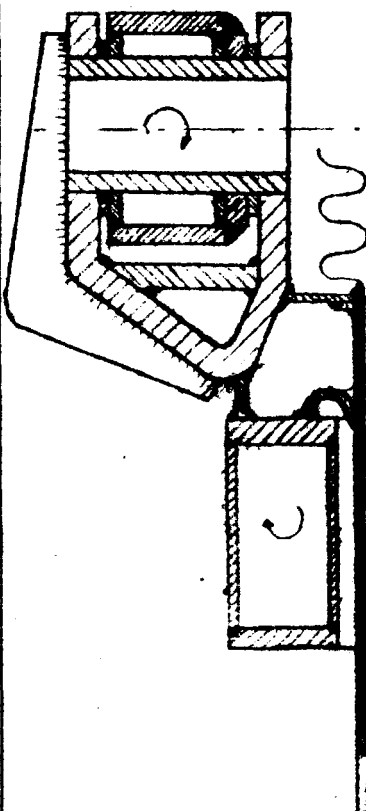
SUBJECT: NASA - Gimbal joint -  
optimization

BY: \_\_\_\_\_

The 4 outside-reinforcement-plates are no longer needed. The weight-reduction is  $(134/145) \approx 10\%$  (see p. 13). The stress-reduction gives little in safety. The new fork-like lug will be heavier however. As an other result the ring-dia may be reduced by 0.8 in. Weight-reduction arising here-from:  $\approx 5\%$ . Serious disadvantages are: Reduction of stiffness if system is not pressurised; considerable increase in axial length.

The lug-cone solution isn't satisfactory. The construction isn't rigid enough, the additional tensile element is very poor. A better construction must be found:

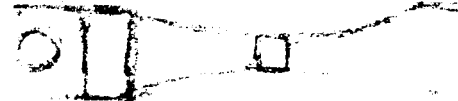
In the following solution the cone is replaced by a ring. This ring is loaded as follows:



It is obvious therefore, that the stresses are smaller than in the case of the ring being loaded at 4 points like the gimbal-ring. Because of the negative eccentricity the flange-ring must be over-dimensioned. The load is transmitted from the fork-like lug to the flange by a flexible element. Another flexible element transfers the cross-load directly to the tube. The gimbal-ring is loaded as mentioned on p. 14. The safety-factor therefore is higher than calculated on p. 8.

flexible compared to ring.

Play - weight may be reduced between pins by machining



## SOLAR AIRCRAFT COMPANY

SUBJECT: NASTI - Gimbal joint -  
optimization  
 BY: W. J. J. J.

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Material - substitution - Gimbal

The gimbal joint of Eng. 38770 is made out of Inconel 718 with a yield strength of 135000 psi (see report M 1697). If the parts are made out of titanium 5AL-2.5 Sn with

184,000 psi	yield strength	} at -320°F
186,000 psi	UTS	

there is a considerable reduction in weight. However, the safety factor must be increased, compared to the original ring, because titanium has no additional safety margin between yield and UTS:

$$\frac{186,000}{135,000} = 1.38$$

$$\frac{1.38 \cdot 1.1}{1.4} = 1.08 \quad (\text{NASTI-factors : 1.1 for yield, 1.4 for UTS})$$

The stresses in titanium-made parts may therefore be 8% higher than the original stresses. This increase is very small. If it is considered, that the stresses due to bend are:  $\sigma = \frac{\text{moment}}{I/c} = \frac{\text{moment}}{BH^3 - bh^3} \cdot 6H$  it is obvious, that the dimensions must remain the same. The same is valid for torsion. Only parts with pure tension could be reduced by  $\frac{1}{1.08}$ . There are no such parts.

Conclusion: The gimbal joint from Eng. 38770 may be made out of titanium without any alterations in dimensions. The reduction in weight is found from the ratio of the specific gravities:

The new joint weighs  $\frac{4.5}{7.8} \left( \frac{\text{g/cm}^3}{\text{g/cm}^3} \right) = 0.58$  times as much as the original one if all parts are replaced. Weight-reduction ~ 42%

## ENGINEERING REPORT

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Material - substitution. BellowsThe following assumption is made:

The bellows can be manufactured out of titanium without reducing the properties of the material.

In the SOLAR - progress - report No 2, the endurance-limit is given for titanium at  $-320^{\circ}\text{F}$ :

Parent Metal : 120000 psi = 65 %

Weld : 80000 psi = 44 %

In a Huntington - Alloy report of Sept. 1963 on Inconel 718 the fatigue-strength is given for  $10^7$  cycles at Room Temp.

min. stress = - max stress : min stress = 0 :

 $S_{\max} = 40000$  $S_{\max} = 68000$  $S_{\min} = 0$  $S_{\min} = 24000$ The bellows from Dwg. 38770 are not welded, so I may take 120 000 psi for titanium. Mr. Psichogios assumes that this is the value for  $S_{\min} = -S_{\max}$ . This value therefore has to be compared to 40 000 :

$$\frac{120000}{186000} = 0.65$$

$$\frac{40000}{135000} = 0.3$$

The excellent behavior of titanium is obvious. The assumption is therefore made that it is superior to 718 for any number of cycles.

The following results are taken from Engr. Report 11-1697 12.90:

Max equiv. pressure stress : 59500 psi (line  
(for bellows) 54000 psi proof  
(Dwg. 38770) 15000 psi limit

alternating flexural stress : 112100 psi

Conclusions: The high pressure stress suggests not to reduce material thickness in case of being made of titanium.  
Weight-reduction therefore is being given here too by the ratio of the specific gravities.

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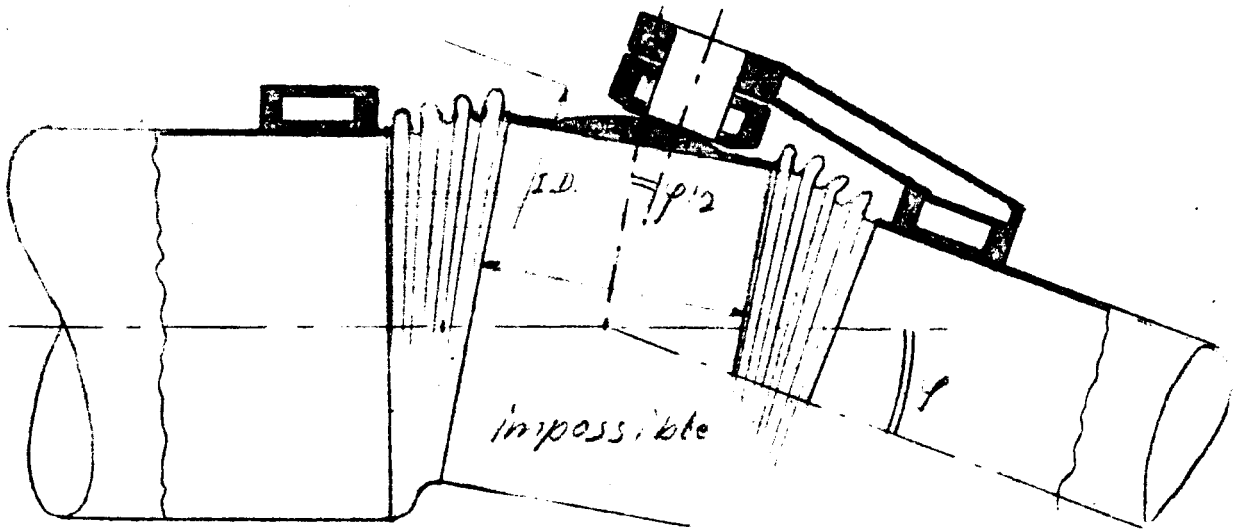
SUBJECT: NASA - Gimmel - joint -  
optimizationDATE 10-28-64PAGE 18 OF \_\_\_\_\_ PAGESBY: O. Tegg

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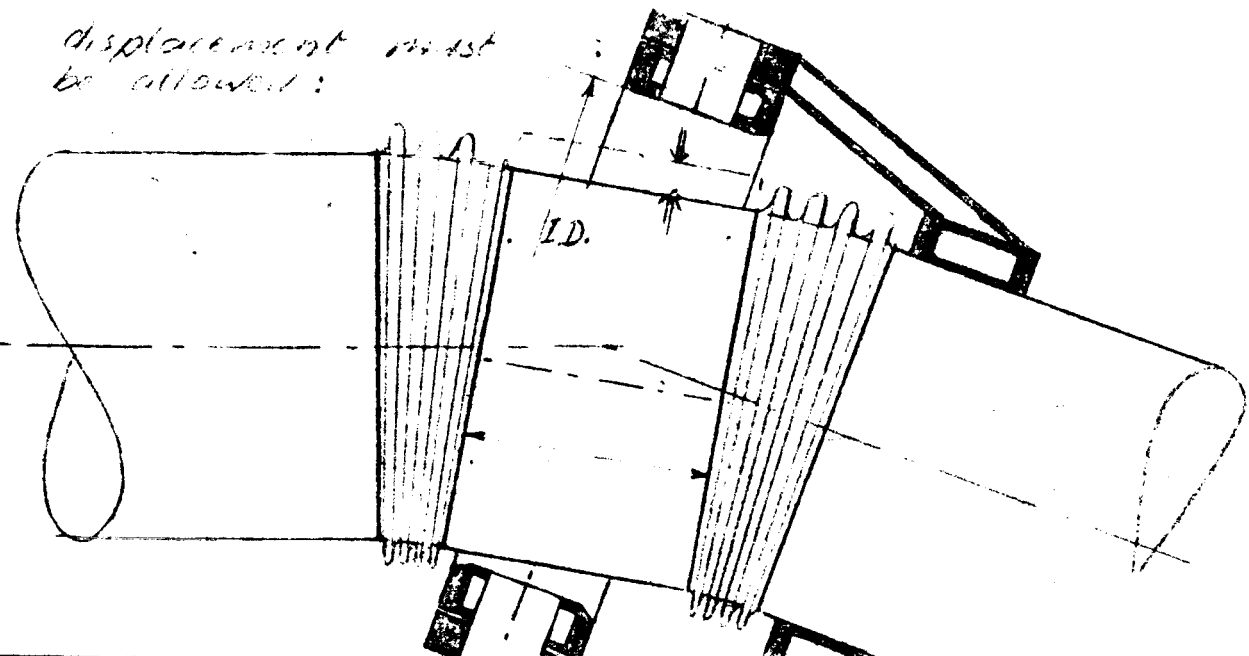
Review of other solutions to joint - problemReduced - Dia - Gimmel - ring (suggested by Mr. Archibald)

The gimmel ring is the nearest part of the joint. The bending- and torsional - moments depend on its dia (see p. 2). If the dia can be reduced, reduction in weight is possible:

Disadvantage: big length needed, bigger bending moment in sqirm - pressure remains same



displacement must  
be allowed:



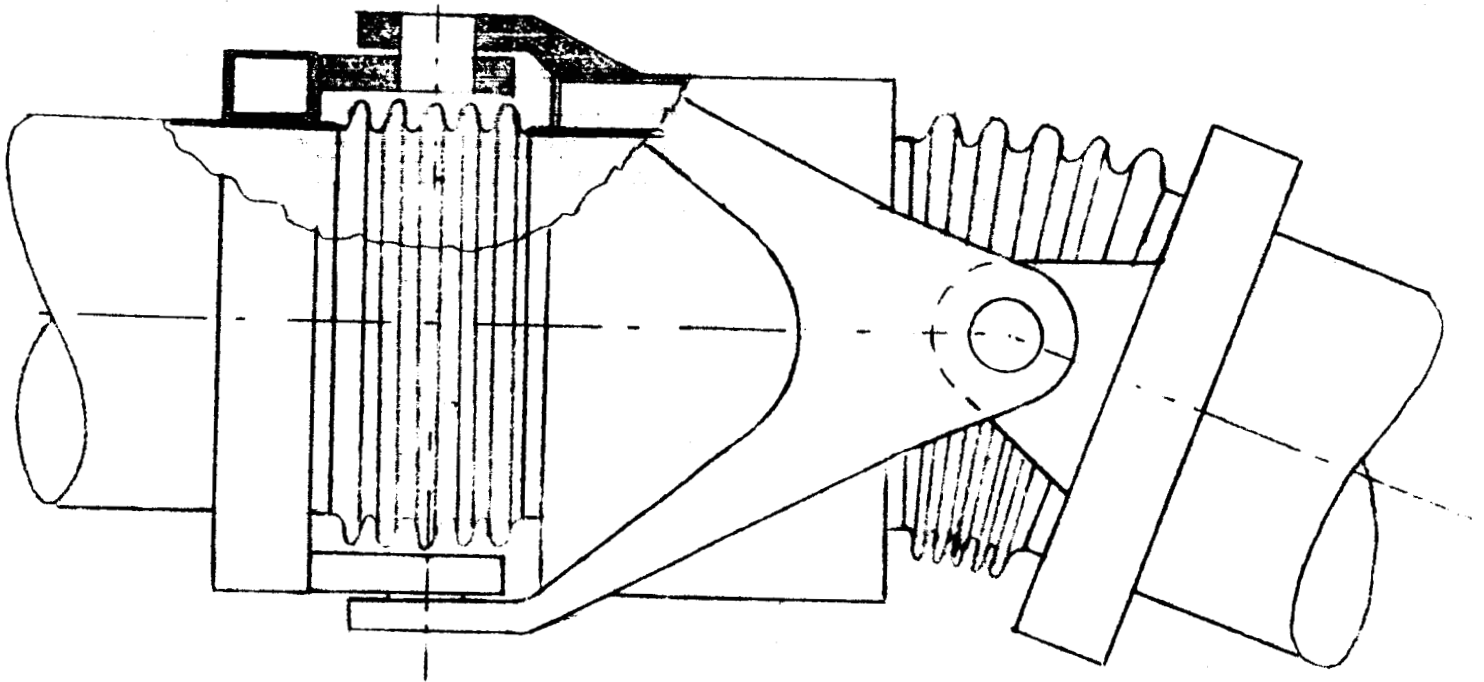
## SOLAR AIRCRAFT COMPANY

## ENGINEERING REPORT

SUBJECT: NASA - Gimbal-joint  
optimisation  
 BY: O. Hegg

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Divide gimbal-joint into 2 simple joints (my suggestion)  
 A gimbal-ring is not necessary in the case of the simple joint. The gimbal-joint can be replaced by two of these:



Advantages: simple, light-weight

Disadvantages: Restrictions in movement, double amount of constructions needed, small articulations only, load-concentration to two points.



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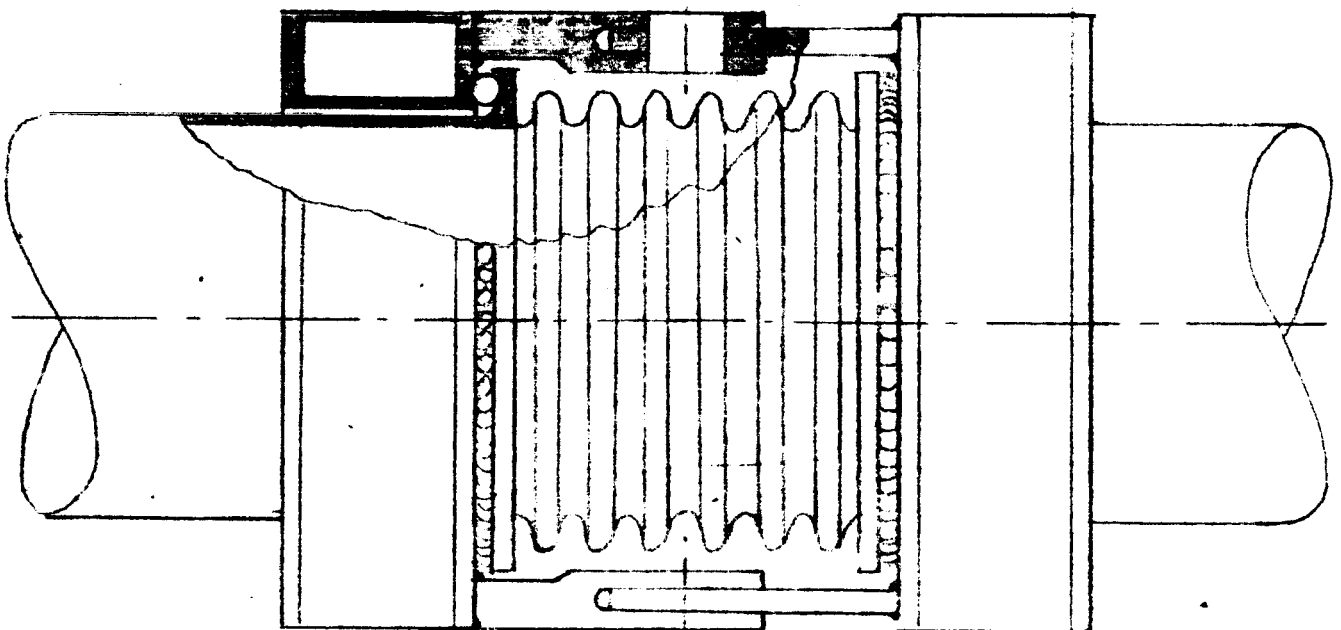
JOB NO. \_\_\_\_\_

SUBJECT: NASA-Gimbal-joint-  
optimisation

BY: C. Hegg

## Rotation-joint (my suggestion)

The gimbal-ring can be eliminated if the flanges may rotate about their axis:



Advantages: no gimbal-ring, light weight

Disadvantages: load-concentration to two points, restriction of movement in one plane, not of non-angled position, ball-bearing needed

But: If the joint is pre-angled there are no restrictions. Pre-angulation can mean that the joint shown above forms an angle in a plane that is normal to this paper. But it can mean too, that the joint is constructed as follows:



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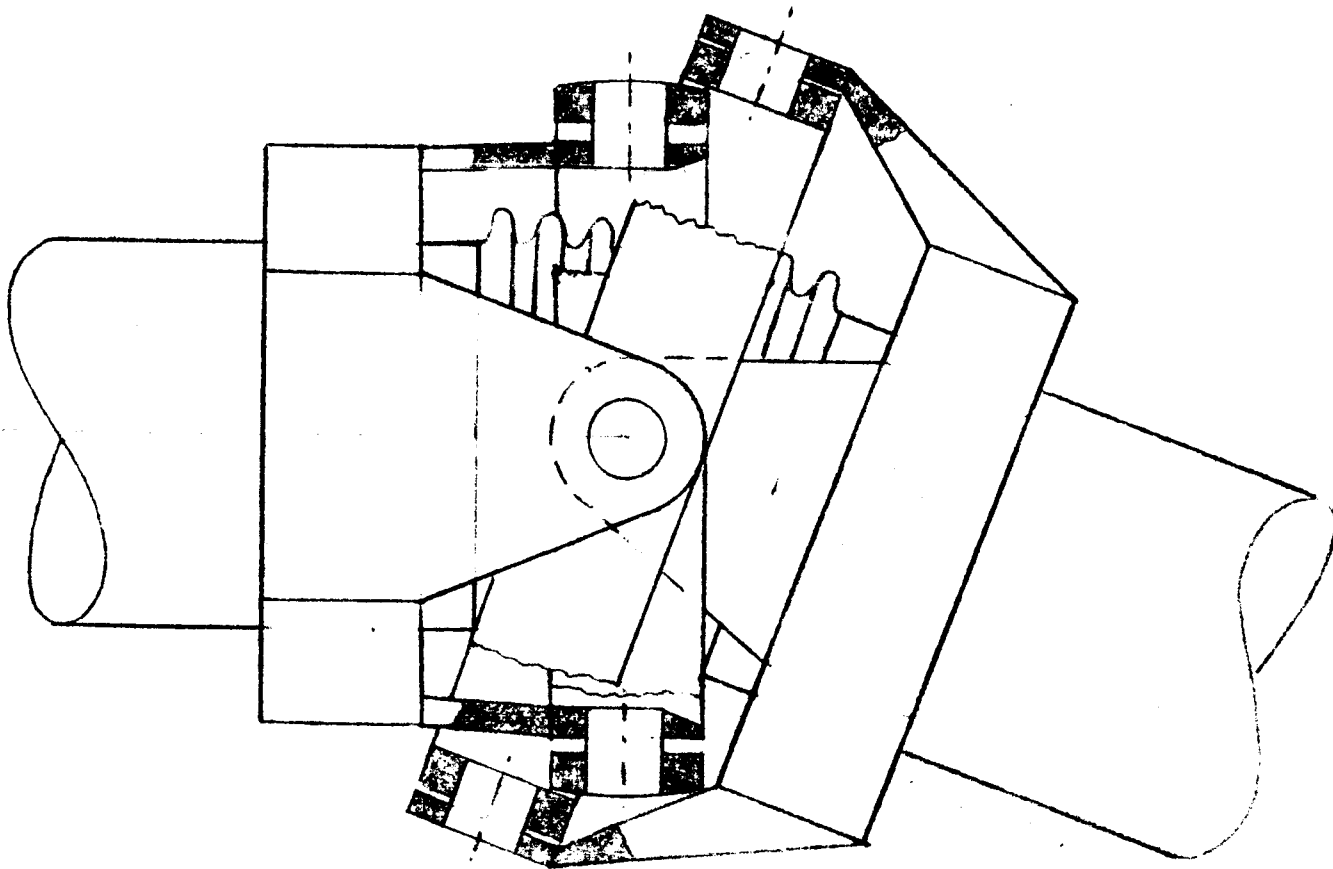
## ENGINEERING REPORT

SUBJECT: NASA - Gimbal-joint -  
optimization  
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### Double ring-joint (suggested by Mr. Michel)

An obvious disadvantage of the gimbal-joint is the load-concentration to two points. The following solution provides a load distribution to four points instead of two:



Advantage: Load-distribution to 4 points on tubes

Disadvantages: Complicated, big outside dia., rings on tubes loaded with very high neg. eccentricity

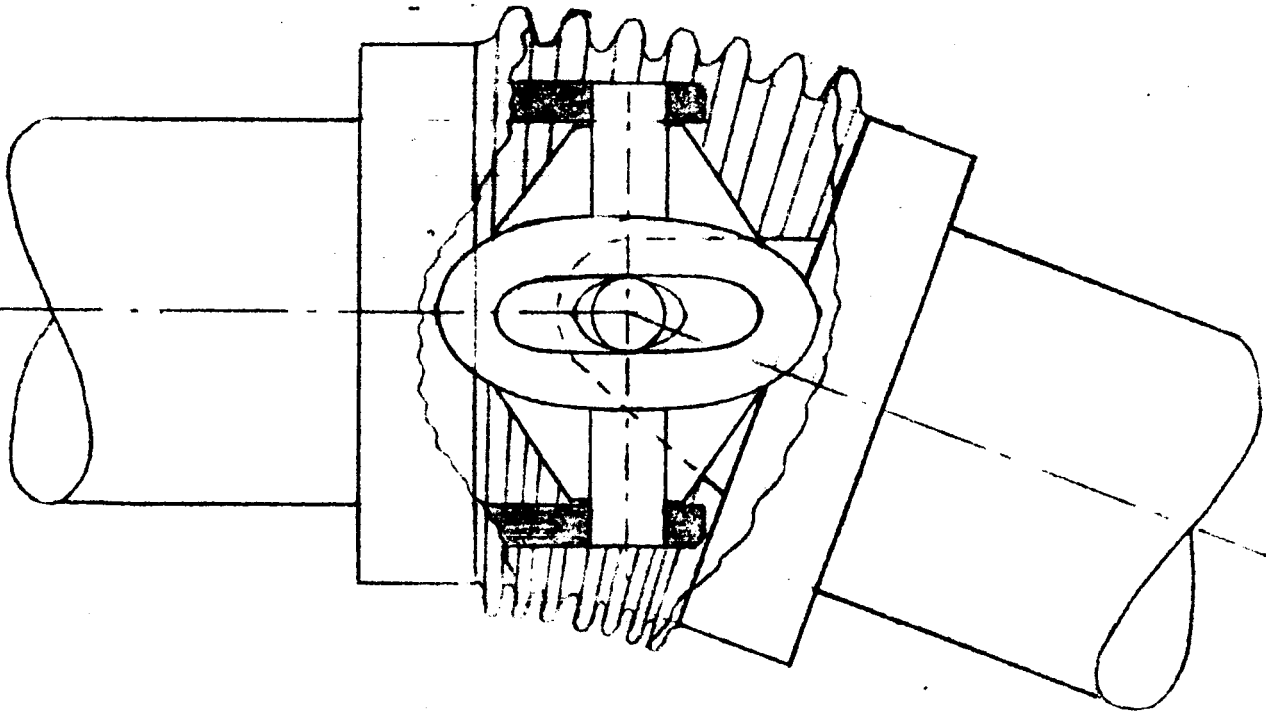
## SOLAR AIRCRAFT COMPANY

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SUBJECT: NASA - Gimbal-joint -  
optimisationDATE 10-28-64BY: J. HeggPAGE 22 OF \_\_\_\_\_ PAGES

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Pin-joint (Existing)

Advantage: Gimbal-ring, can be replaced by light-weight cross-shaped structure.

Disadvantages: Small angulations only (cross does not turn in direction of stream-lines), load-concentration to two points, increased bellows-dia., pressure-drop

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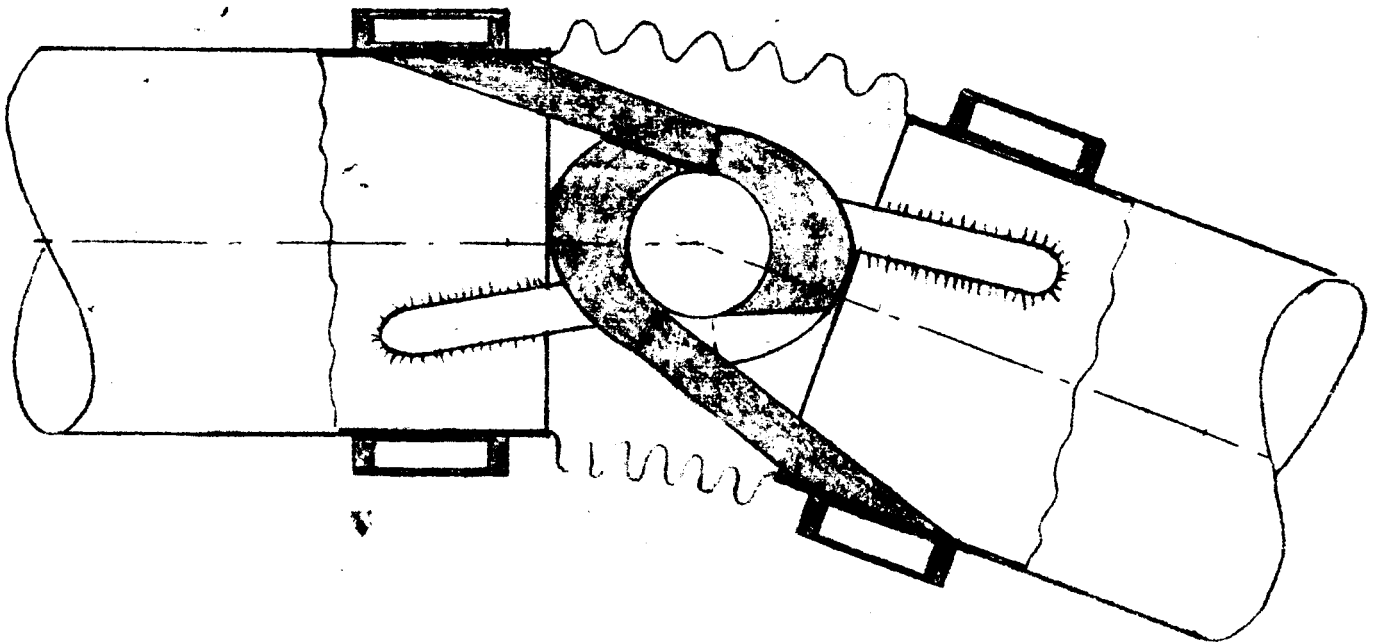
## ENGINEERING REPORT

SOLAR AIRCRAFT COMPANY

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 BY: D. Heege

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Internal ball-joint (existing)



Advantage: load-distribution to 3 or 4 points possible  
 Disadvantages: small angulations only, structure cannot be  
 stream-line-formed, pressure-drop.

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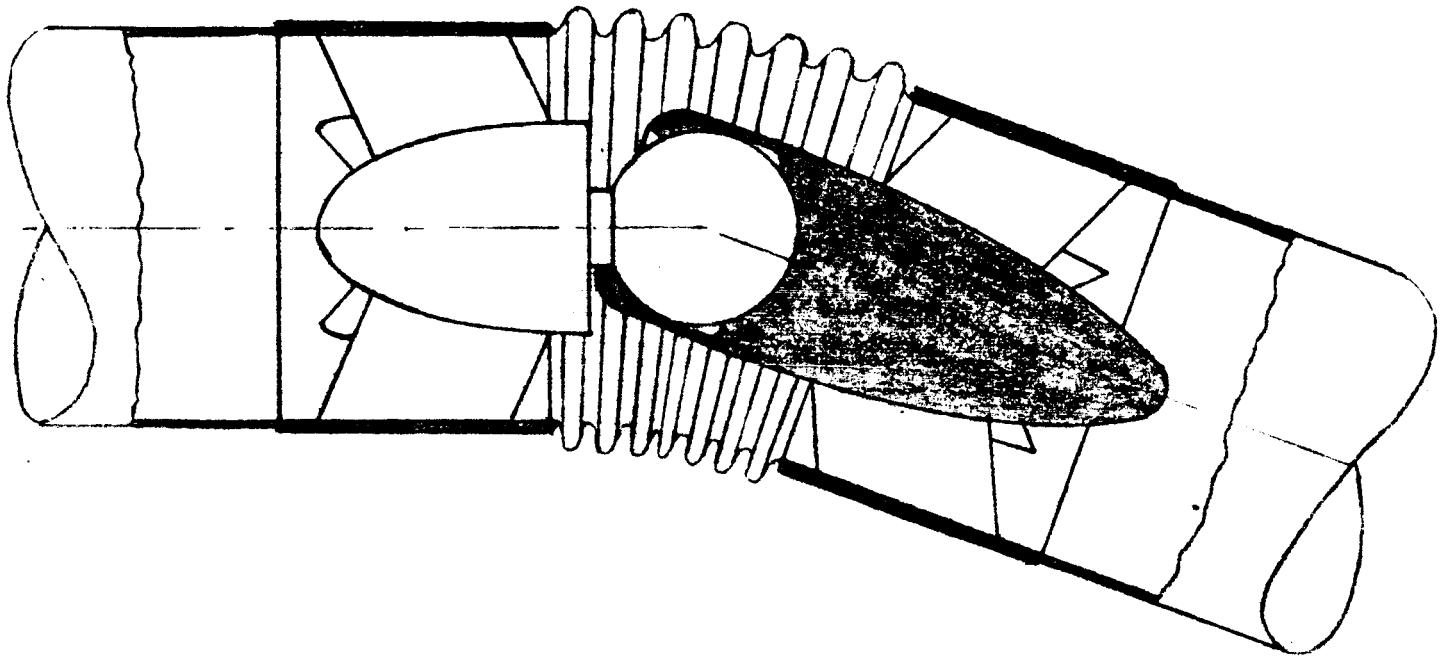
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SUBJECT: NASA-Gimbal-joint-  
optimisation  
BY: J. Tegg

Internal - ball-joint (existing)



Advantages: load-distribution on many points  
stream-line-inside with low pressure drops.

Disadvantages: small angulations only (for small ball)  
no torsion transmitted

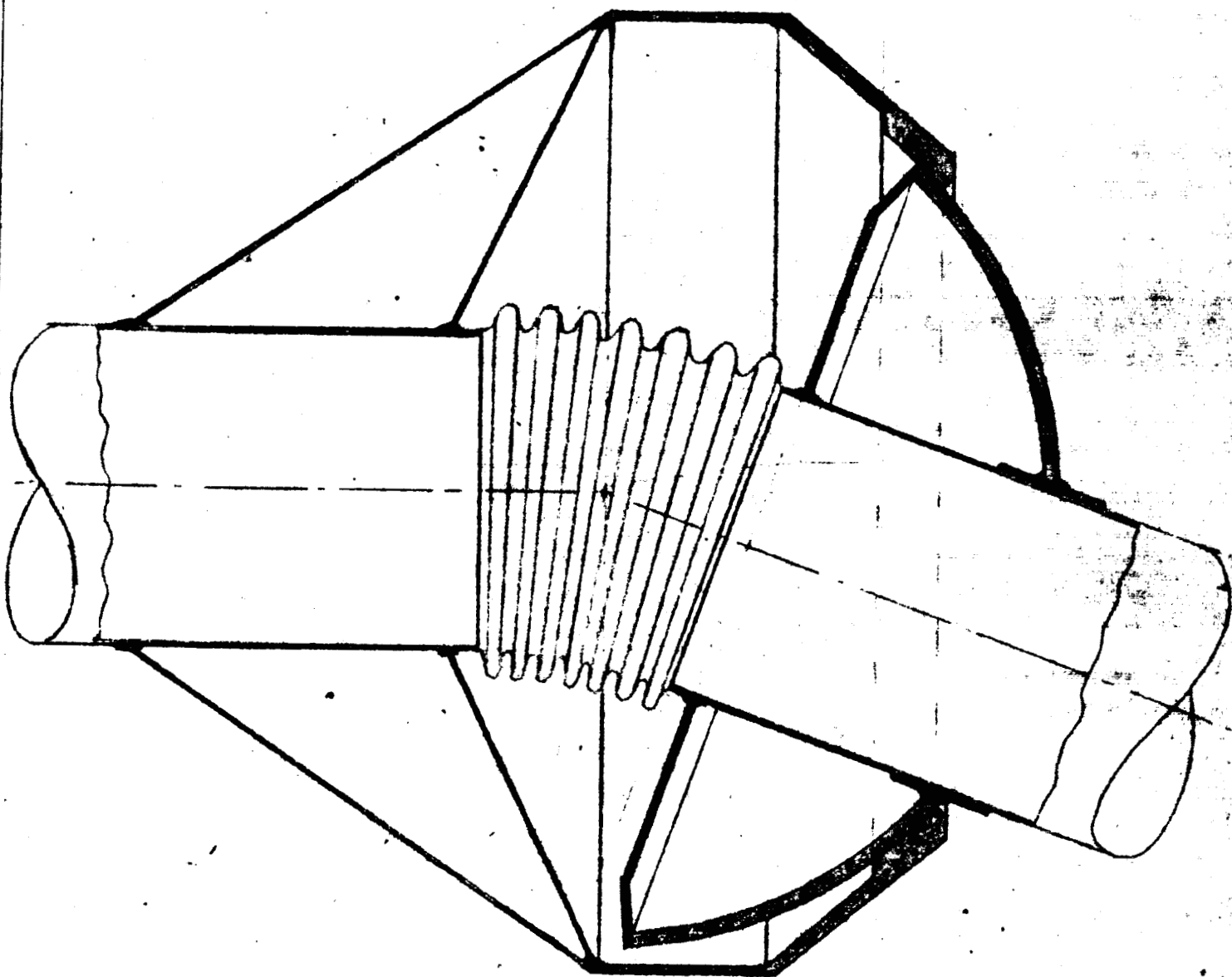
## SOLAR AIRCRAFT COMPANY

## ENGINEERING REPORT

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optimisation  
 BY: C. Webb

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External - ball joint



Advantage: load distribution

Disadvantages: very small angulations only, clumsy,  
 great angulation forces, no torsion trans-  
 mitted

# SOLAR



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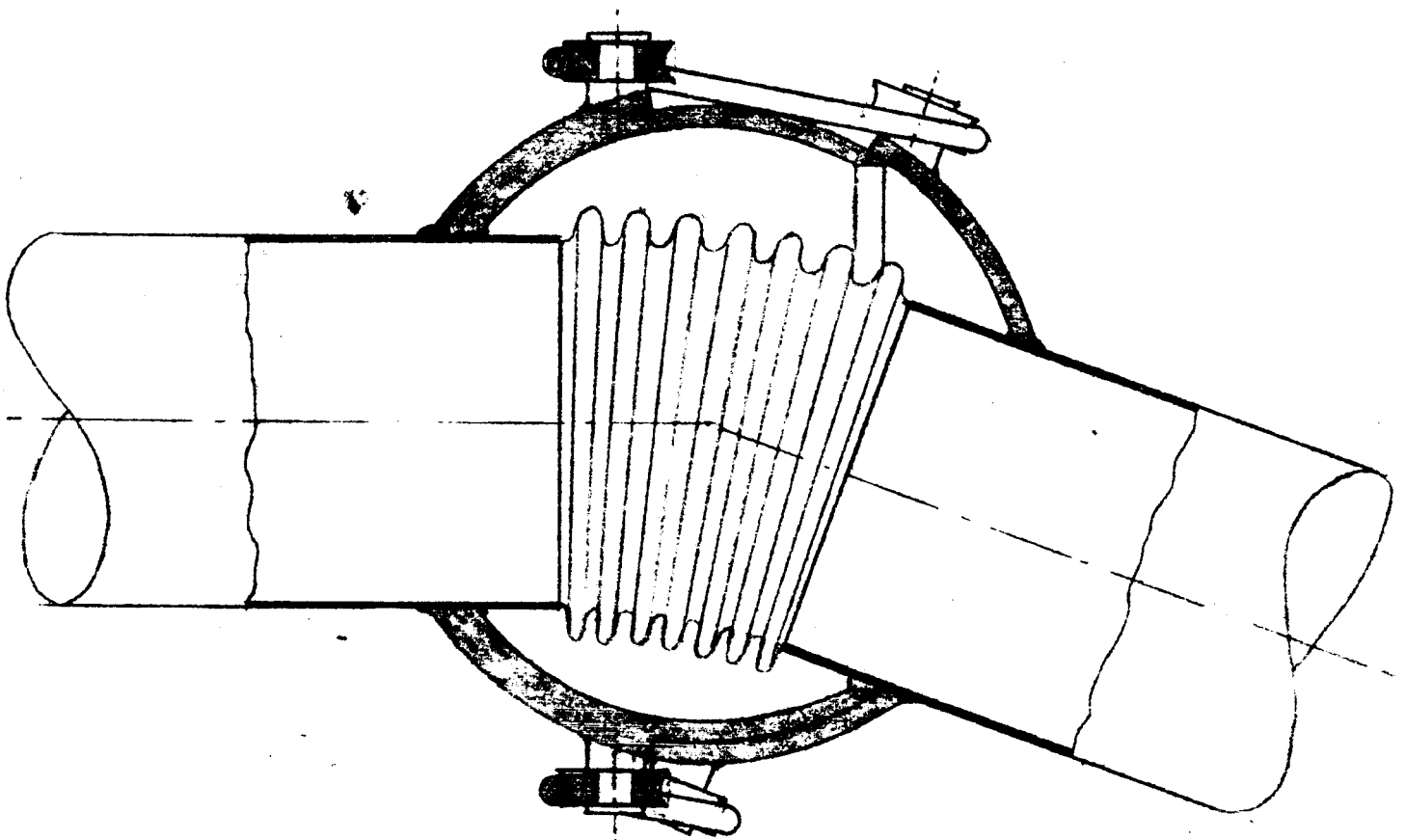
JOB NO. \_\_\_\_\_

SUBJECT: NASA - Wind Tunnel Joint

2. Wind Tunnel - 2000

BY: P. H. [unclear]

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*Advantage: Load-distribution*

*Disadvantages: complicated, low pressure, small angle,  
big angulation force.*

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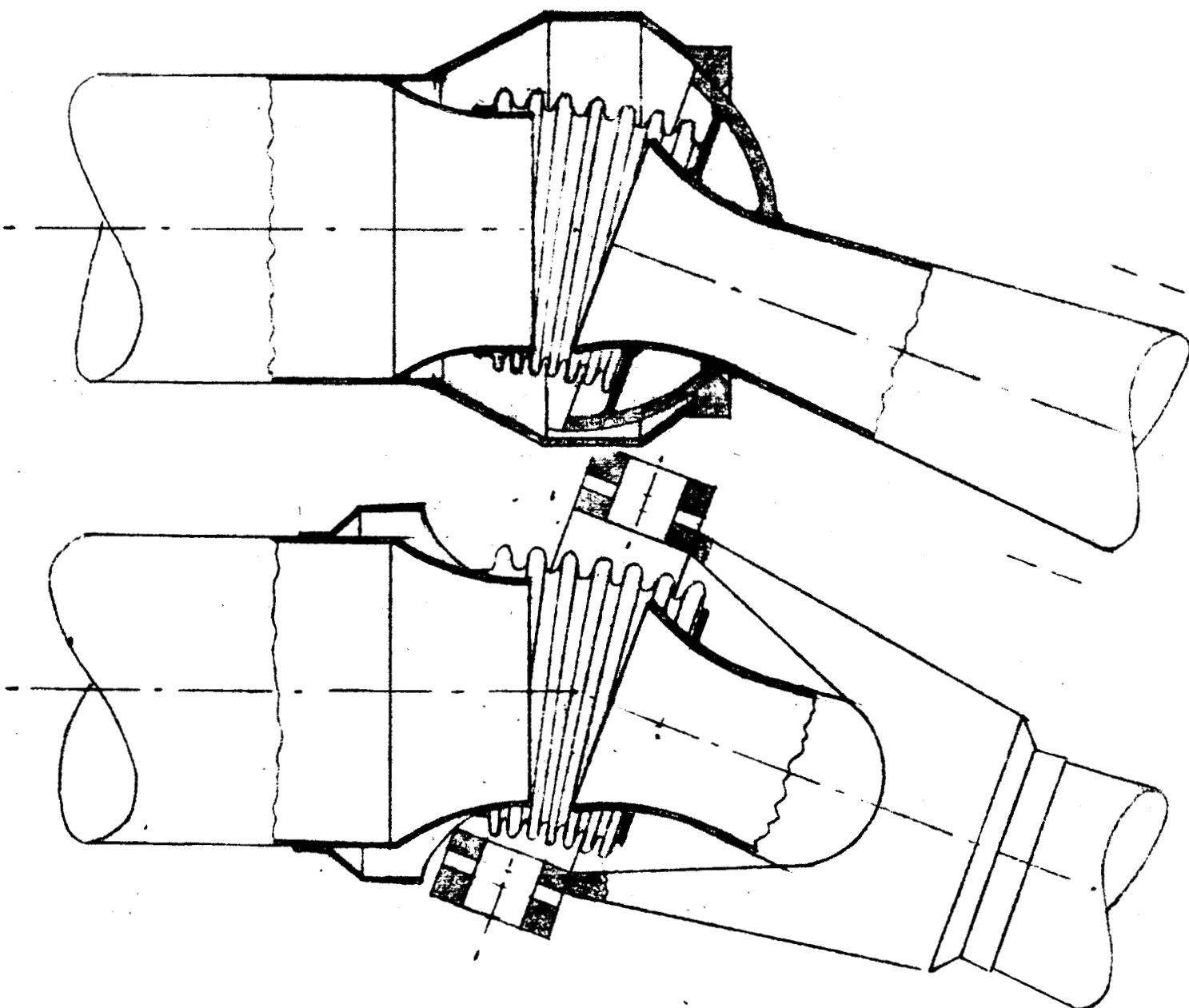
SUBJECT: NASA - Gimbal-joint - optimisation

BY: D. Fogg

Venturi-reduced-dia-joint (my suggest. to use with ball-joint)

Advantages: compact (see also next page), light-weight, use with gimbal-joint or ball-joint, small bellows, bellows may be externally pressurised (see p. 32)

Disadvantages: bigger angulation forces, big length, pressure drop



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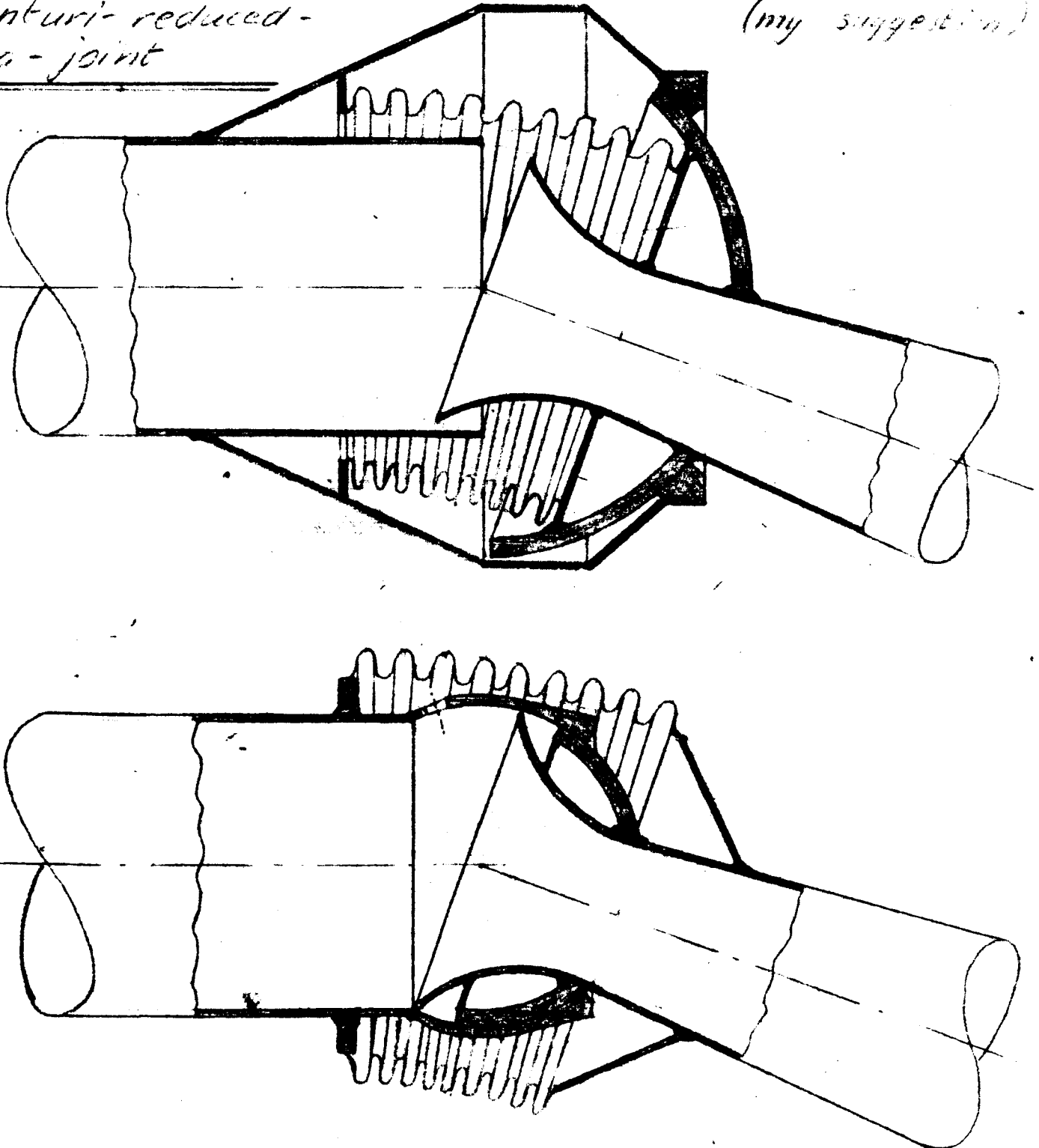


SUBJECT: NASA-Gimbal-joint-  
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Venturi-reduced-  
dia-joint

(my suggestion)



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SUBJECT: NASA - Gimbal - joint -optimizationBY: D. H. G. G.

## Result

The usual gimbal joint is only one possible solution. There are other possibilities which offer great advantages. The solutions on p 18, 20, 24 and 27/28 seem to be worth to be further investigated. Some additional sketches showing these joints with reasonable dimensions and some calculations will therefore be made to show proper dimensions and approx. pressure drops.



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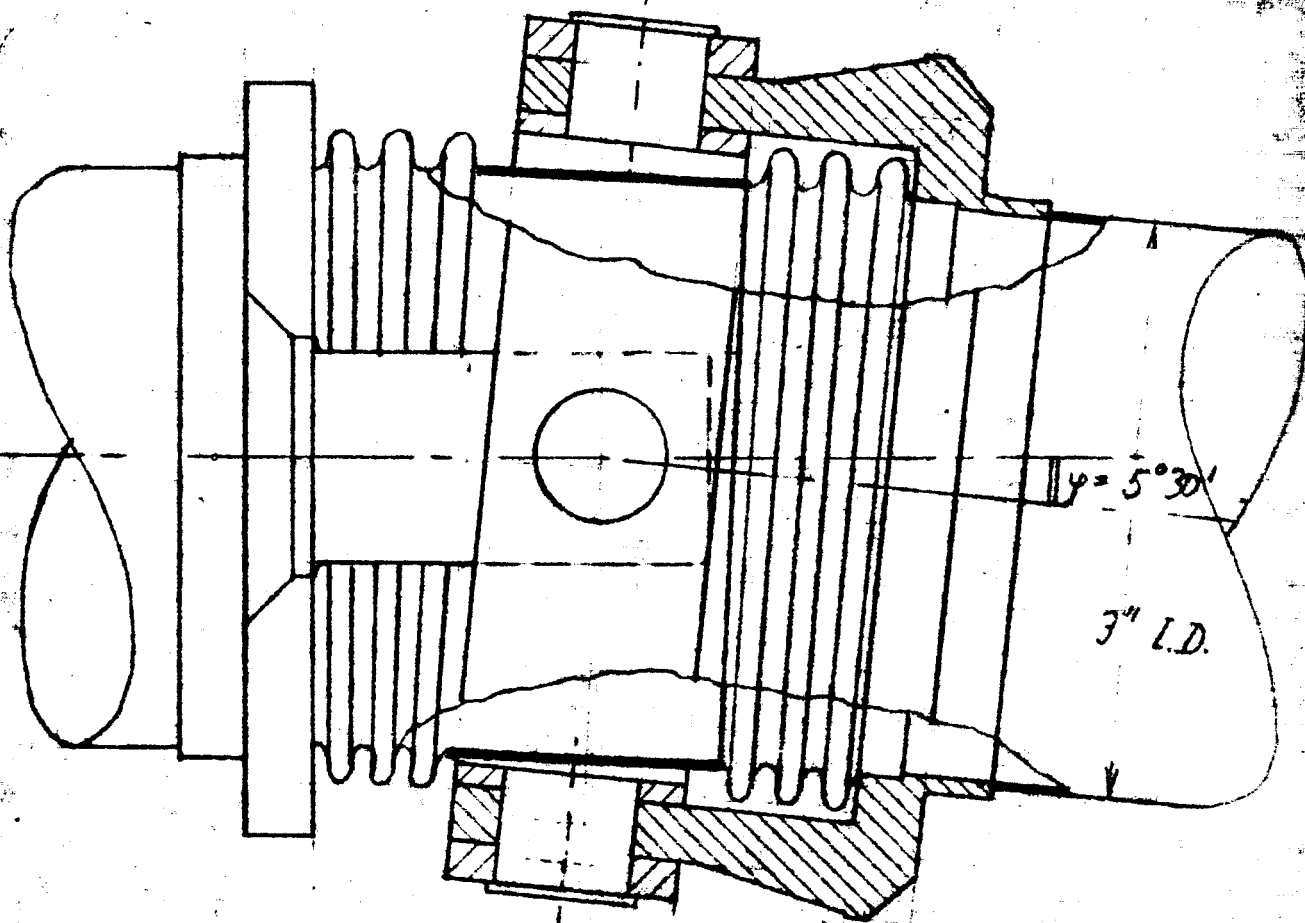
SUBJECT: NASA-gimbal-joint -  
optimisation

BY: O. R. Bagg

## Reduced-dia-gimbal-ring

Reduction of ring-dia: old O.D. : 5" } reduction 10%  
new O.D. : 4.5"

The lugs must be stronger because of the greater bending moment resulting from the across-load.



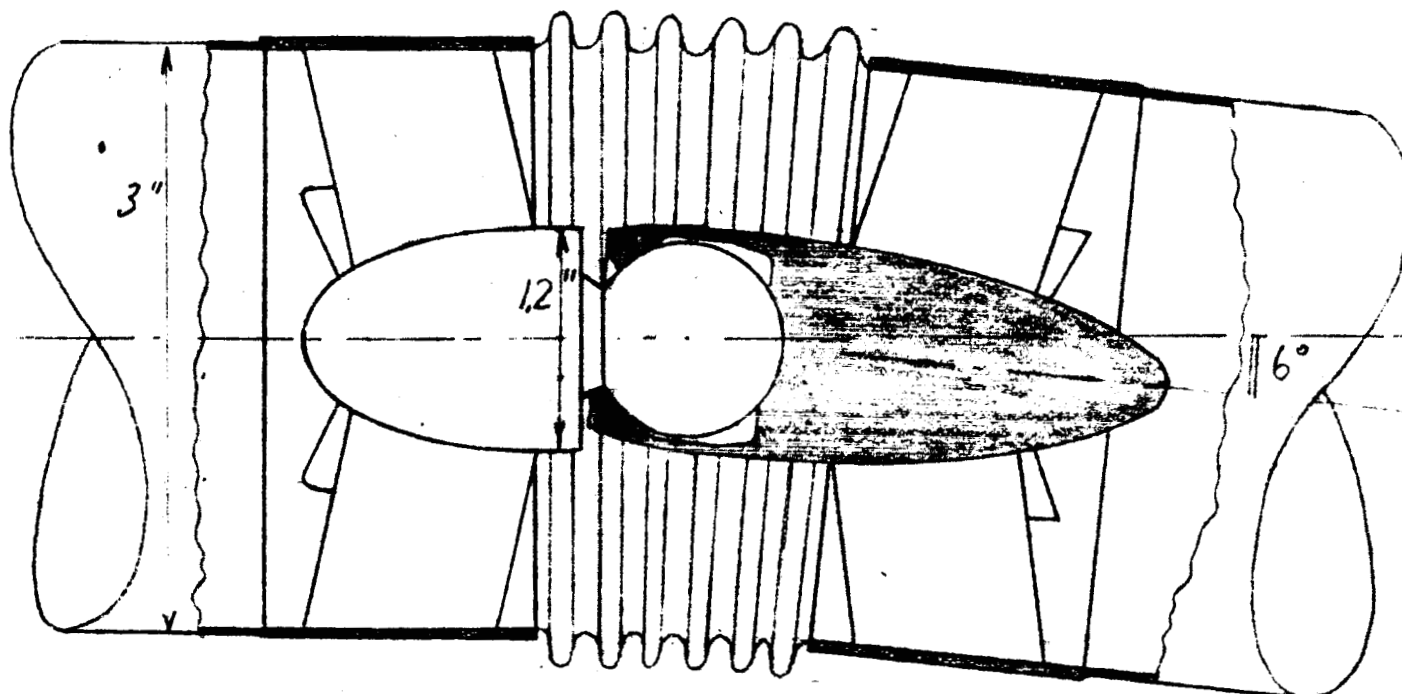
The flanges may remain the same; they have to take the bigger bending-moments from the across load but the basic strain is smaller (lugs closer to center).  
The squirm-behaviour of the bellows isn't improved.  
Light-weight hollow profile cannot be used.



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## Internal-ball-joint



Smallest solution. Tensile stresses at burst: 90 000 psi  
 Bearing pressure " operating: 24 000 psi  
 Reduction of flow - cross-section:

$$\left(\frac{1.2}{3.0}\right)^2 = 16\%$$

$$\text{Ratio of velocities: } \frac{1}{0.84} = 1.19 = \frac{W_{max}}{W}$$

Approx. pressure-drop:

$$\Delta p = \frac{f}{2} [W_{max}^2 - W^2] = \frac{f}{2} W^2 \left[ \left( \frac{W_{max}}{W} \right)^2 - 1 \right]$$

$$= \frac{f}{2} W^2 \cdot 0.42$$



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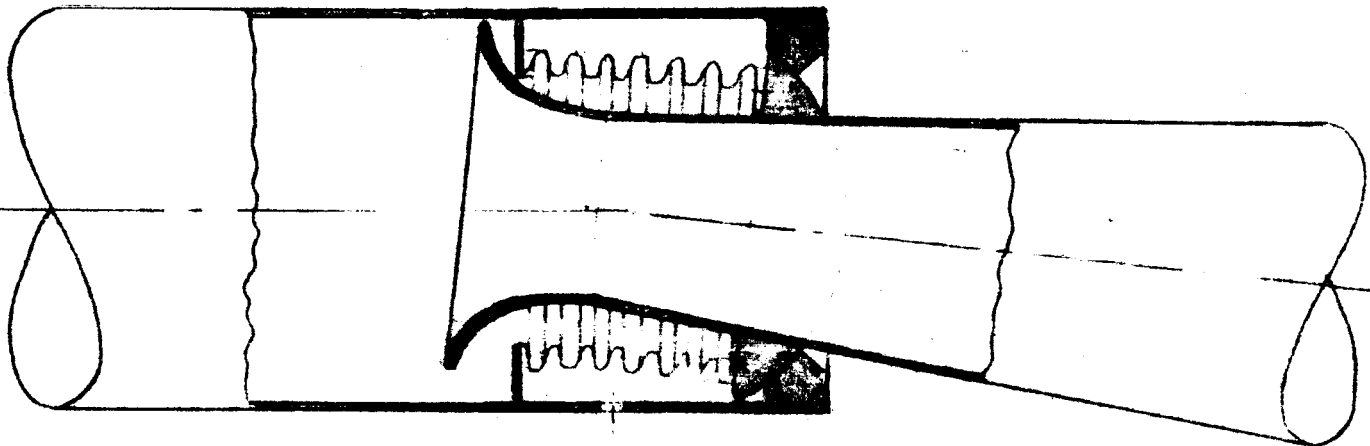
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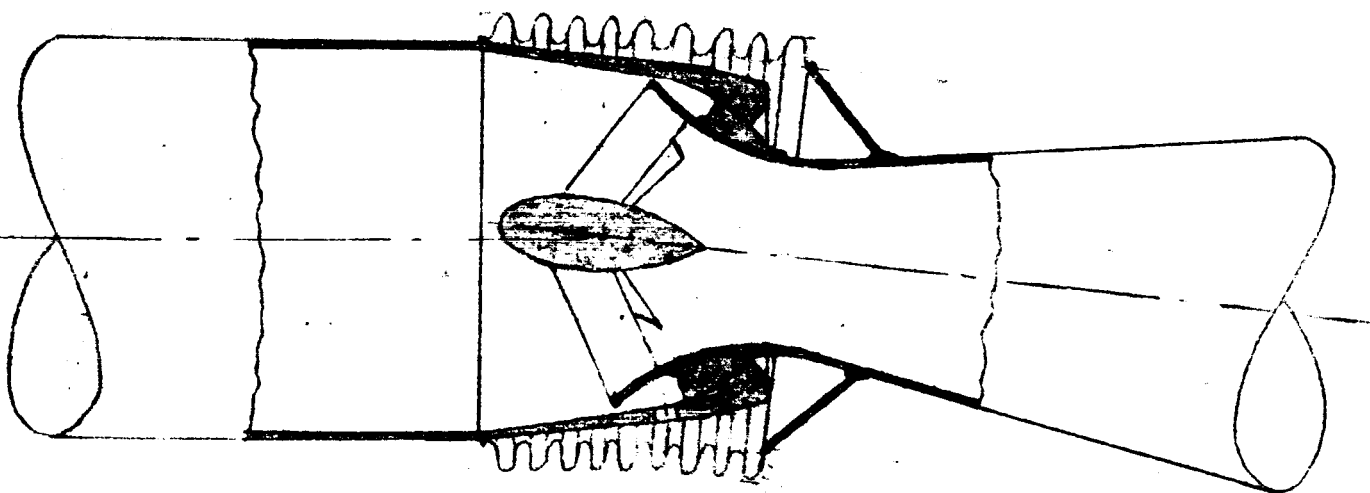
SUBJECT: NASA-Gimbal-joint -  
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Venturi-reduced-dia-joint (my suggestion)



*possible solutions, see next 2 pages*



*guide vanes produce twist,  
allow shorter venturi  
see "Dubbel I" p. 301*



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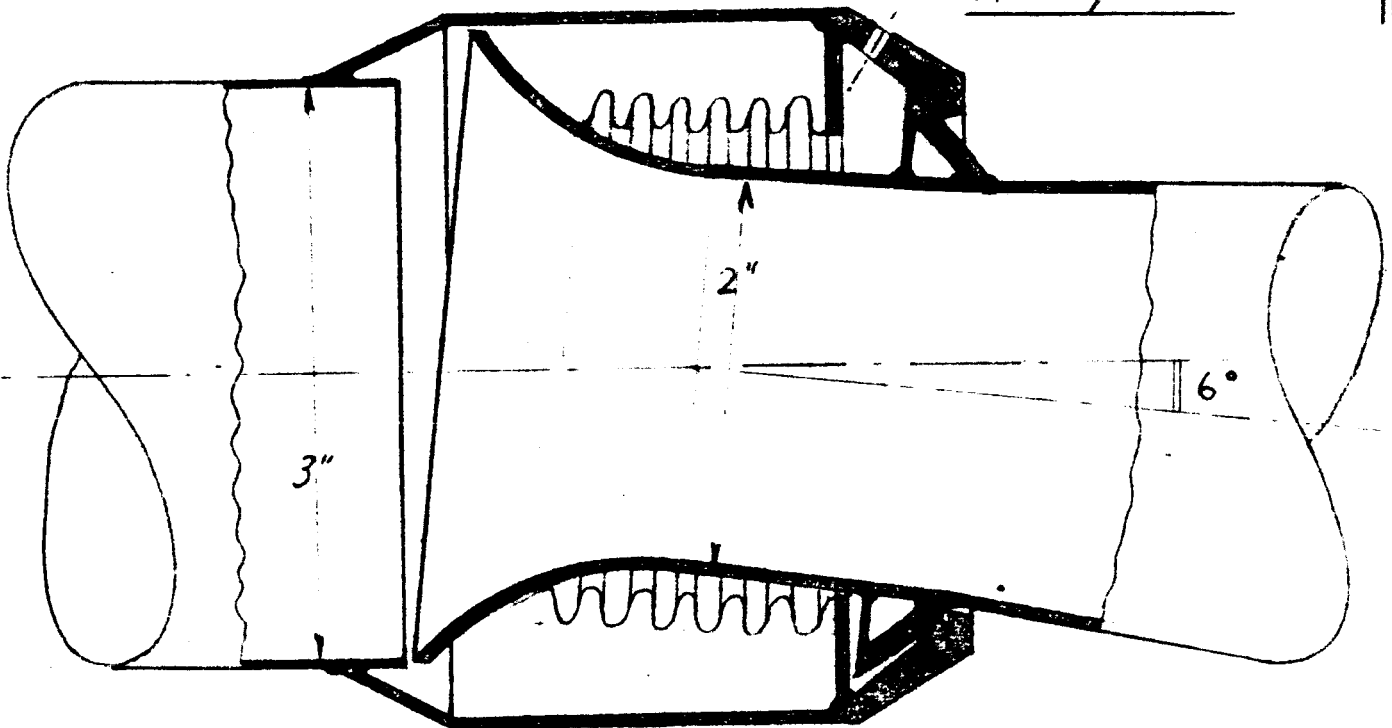
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SUBJECT: NASA-Ginibal-joint -  
optimisation

BY: O. Hegg

Venturi - reduced-dia-joint

bellows outside-  
pressurised!  
no squirm!



*solution with smallest weight.*

*Absolutely equal load-distribution.*

*Ratio of velocities:  $\frac{W_{max}}{W} = \left(\frac{3}{2}\right)^2 = \frac{9}{4} =$*

*Pressure drop:*

$$\Delta p = (0.1 \dots 0.25) \frac{\rho}{2} [W_{max}^2 - W^2] = (0.1 \dots 0.25) \frac{\rho}{2} W^2 \left[ \left(\frac{9}{4}\right)^2 - 1 \right]$$

$$= \frac{\rho}{2} W^2 \cdot (0.40 \dots 1.0) \quad (\text{see "Dinoball 2" p. 300})$$

*Length of joint:  $7.5" = 2\frac{1}{2} D$*



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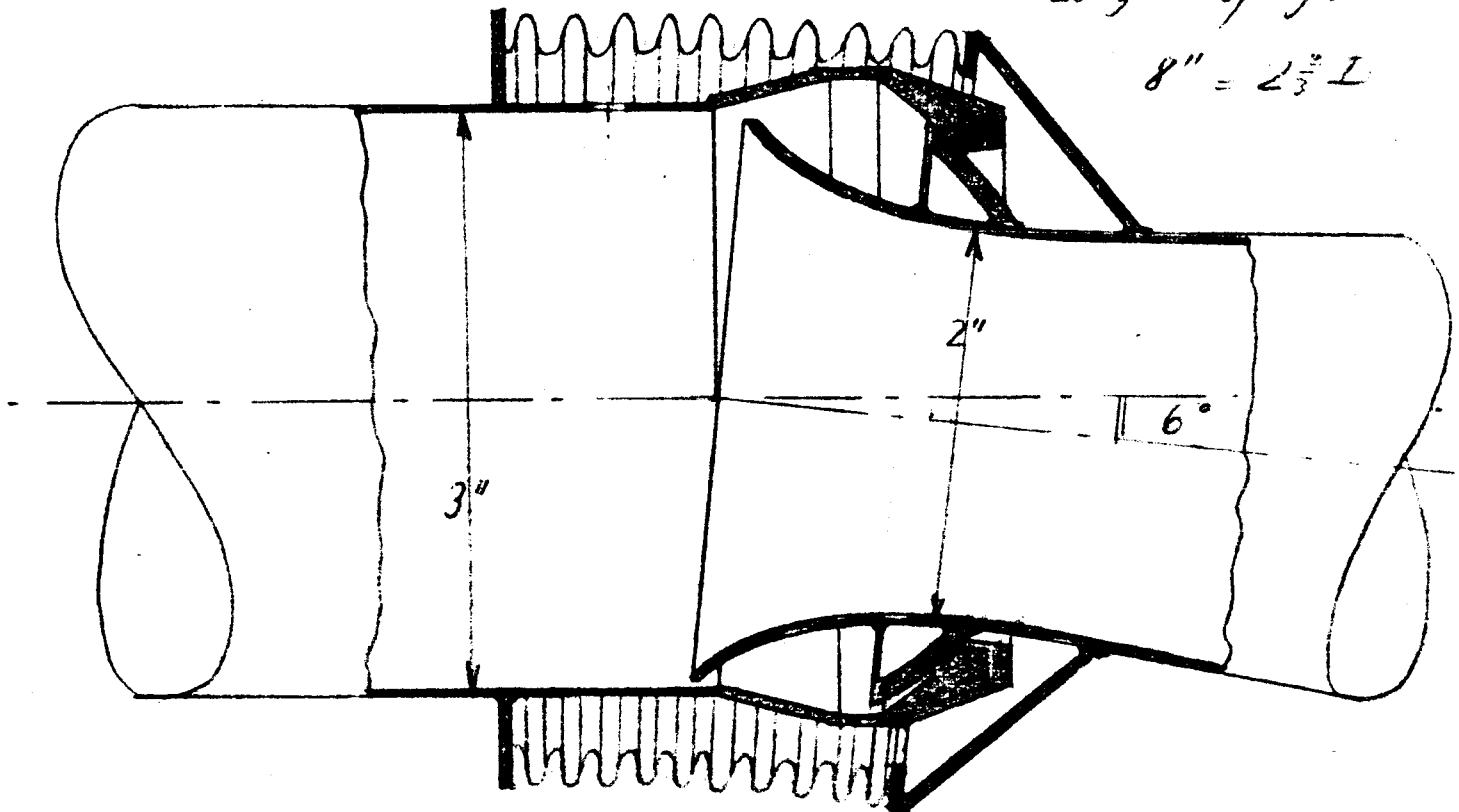
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SUBJECT: NASA - Gimbal-joint -  
optimization  
 BY: D. Hagg

## Venturi-reduced-dia-joint

Length of joint:

$$8" = 2\frac{2}{3} I$$



The same remarks of p. 32 are valid here.

It should be added too, that convergent flow-sections are generally used at many places in angulated ducts to reduce pressure losses, for example in suction-pipes of Francis- or Kaplan-water-turbines or in air-ducts:





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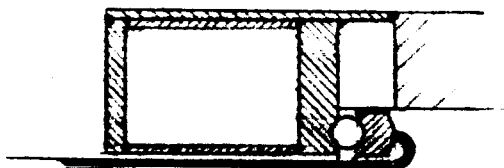
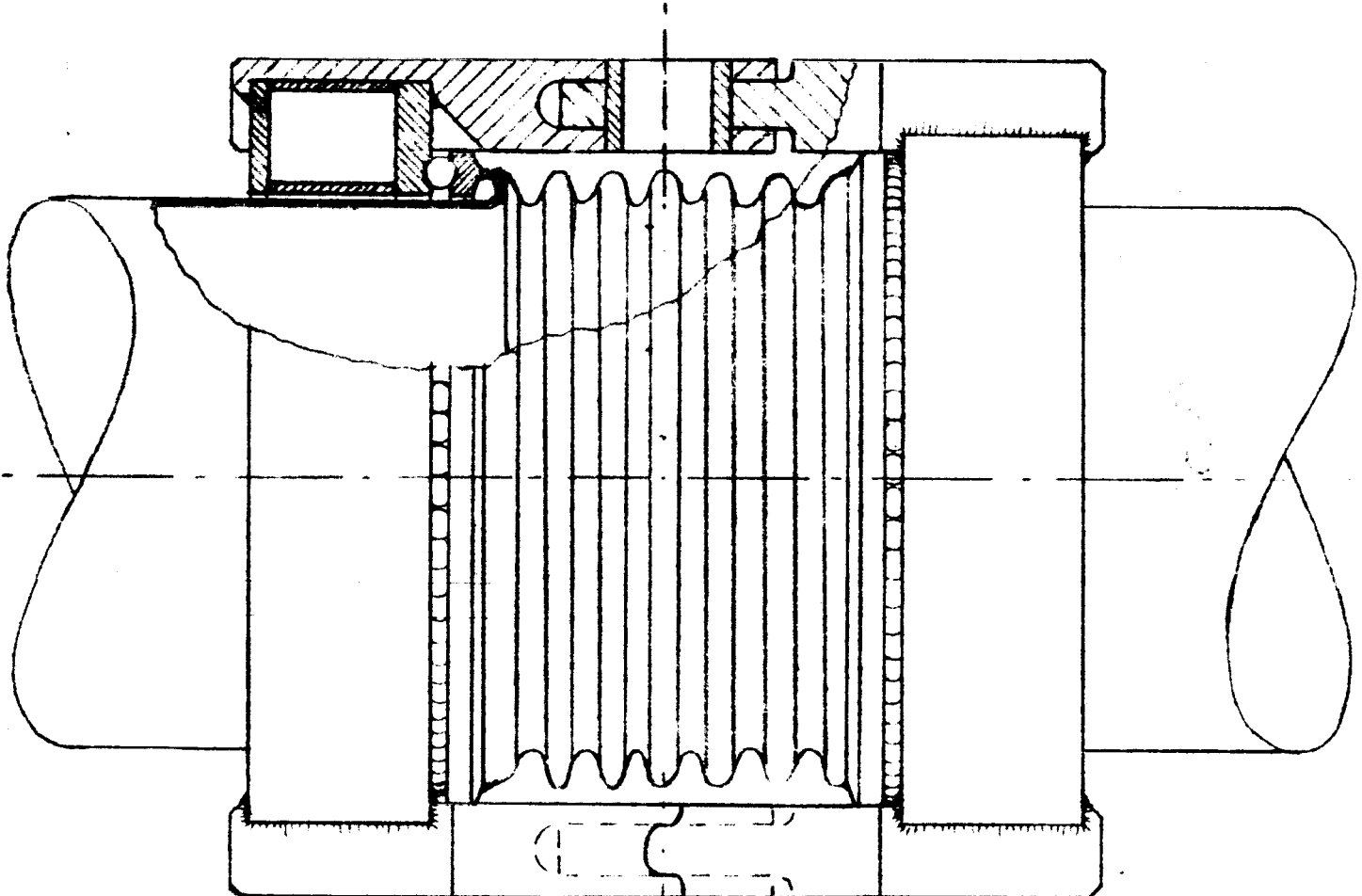
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Rotation-joint (my suggestion)



Detail of other construction

*This joint could easily be combined with the venturi-reduced-dia-joint for weight-reduction and smaller dimensions. Outside - pressurised as in p. 33 squirm would be eliminated.*



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BY: O. Abegg

## Discussion

In the general case a gimbal-joint may be used. However, under certain circumstances other solutions must be considered:

### A. No Flow-restrictions permitted:

1. If no angulation-restriction is permitted the simple gimbal-joint (p.15) may be replaced by the reduced-dia-gimbal of p.30 for weight-reduction. Small angulations only.
2. Rotation-joint p.35 weighs far less; suitable for big angulations, angulation-restriction in one plane.

### B. Flow-restrictions permitted:

1. If pressure-drop shall be minimum the internal-ball-joint seems advantageous (p.31) size is absolutely minimum.
2. If higher pressure-drop (see p.33) is allowed the venturi-reduced-dia-joint p.33/34 must be chosen. Higher angulation-forever. Weight is absolutely minimum.

REF.: MACHINE DESIGN, OCT. 15, 1959 P 147-155